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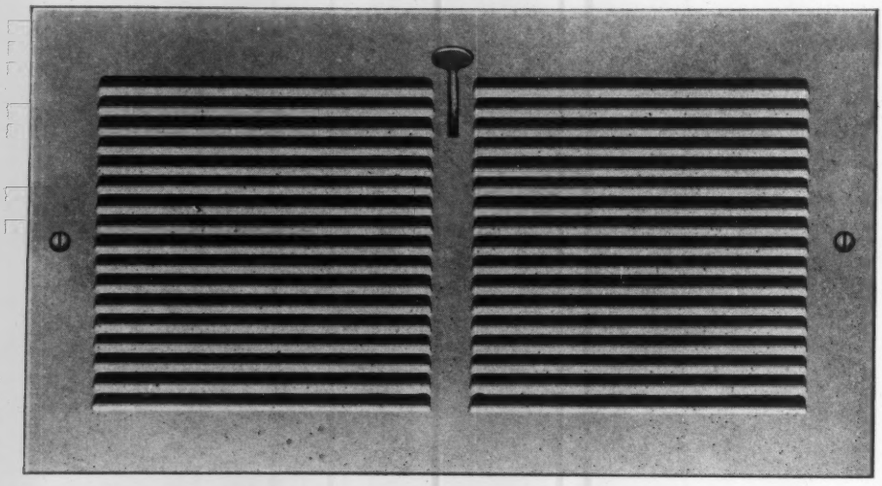

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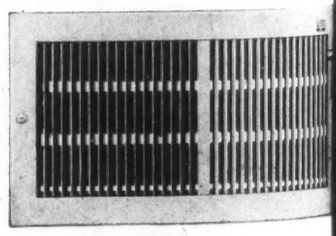

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The Editor's Pages

SKYSCRAPERS BIG AIR CONDITIONING MARKET

America's skyscrapers—the office buildings, apartment houses, hotels and hospitals 10 stories or more in height—represent the greatest immediate commercial market for air conditioning, in the opinion of Dr. Willis H. Carrier. "The latest figures available show that only 5.46 per cent of the nation's office buildings are air conditioned," he points out. "The apartment house picture is even more startling, with only 0.3 per cent air conditioned. According to a recent survey in the hotel field, 13.9 per cent have been completely or partially air conditioned."

A recent survey showed that air conditioning has been installed in 3010 industrial plants, or 1.78 per cent of the nation's 169,111 factories, according to Dr. Carrier.

In the industrial market, tobacco factories rank first as the most completely air conditioned industry. Yet only 25.3 per cent of the nation's tobacco plants are air conditioned.

Candy factories rank second—a survey of the nation's candy plants, conducted last summer, revealed that 14.4 per cent of the total had air conditioning in one form or another. Much of it had been installed more than 10 years ago, however. The candy industry, because of its chocolate dipping problem, was one of the first to recognize the importance of air conditioning. In third place in the industrial field are fur storage plants, with 6.5 per cent. In only a few other industries has even 1 per cent of the saturation point been reached up to now.

In commercial markets, 40,247 installations have been made. This represents a total of only 2.8 per cent of the saturation point in that field, says Dr. Carrier. In the commercial market, theaters outrank all other markets by a wide margin. Department stores rank second, although only one out of every five department stores has been air conditioned. Drug stores report 1.58 per cent air conditioned; restaurants, 2.3 per cent; and banks, 2.89 per cent, according to Dr. Carrier.

* * *

Called upon to provide shelter and accessories for 1,195,000 troops by June 30, 1941, the construction division, office of the quartermaster general, had housing available for 1,214,000 men on that date, Brigadier-general Brehon B. Somervell, chief of the construction division, told the summer convention of the Carolinas branch of Associated General Contractors of America last month.

General Somervell pointed out that during the past year the construction division has conducted a building campaign of an estimated cost somewhat in excess of $1,500,000,000. This program was carried on in all parts of the continental United States, Puerto Rico, Hawaii and the Panama Canal Zone. It includes about 450 projects on 250 different locations.

* * *

SILVER RESEARCH PROGRESSES

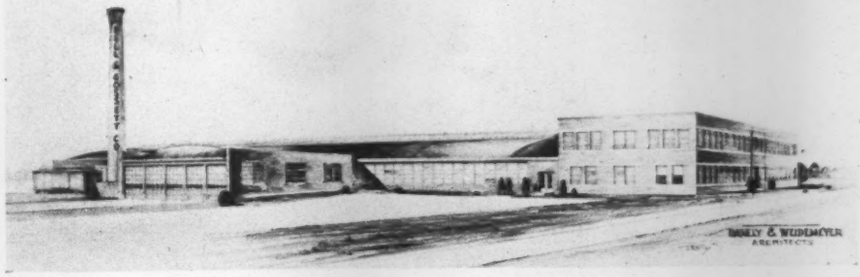
The American Silver Producers' research project, sponsored by several U. S. silver producing companies, has completed a year of activity at the Bridgeport plant of Handy & Harman, according to a progress report issued last month.

Included in the work are experiments to determine the strength of extruded tubing made from a 3.5 per cent silver, 96.5 per cent tin alloy. A bursting strength of 2500 psi was obtained and this is apparently far in excess of any working pressure encountered in distilled water lines where this material is finding commercial use. Tests on threaded joints showed that the alloy had a tensile strength 25 per cent greater than joints made with pure tin tubing, it is reported. For certain installations it would seem feasible to use threaded connections in distilled water lines if the tubing were made of the silver-tin alloy. Both the 3.5 per cent silver and 5 per cent silver-tin alloys are finding applications as solders.

Study has also been started on the corrosion of silver to silver joints made with different silver brazing alloys. This is of particular interest in connection with the manufacture of silver lined chemical equipment.

THEY TAKE THEIR OWN MEDICINE

As might be expected, the new Bell & Gossett factory and administration building at Morton Grove, Ill., is heated by forced circulation hot water, under zone control to allow flexibility of heat distribution and permit full compensation for occupancy. Executive offices are on one zone, the main office on another, the factory on another, etc. Thus, if one or two departments are working overtime their zones can be maintained at the regular temperature while the others drop to a lower temperature . . . . . Various types of heating and radiator units are used and in the engineering department and factory locker and washroom, panel heating with pipe coils in the floor is being installed . . . . . Two oil fired boilers using Bunker C fuel serve the new building; each will have submerged heaters for heating domestic water instantaneously without the use of a storage tank . . . . . The building, made necessary by a large increase in the use of forced hot water heating, is to be ready for occupancy in September



AIR CONDITIONING AND NATIONAL DEFENSE

The following listing of "uses of refrigerating and air conditioning machinery with regard to the effect on the national defense program and the national economy" has been compiled by the Air Conditioning & Refrigerating Machinery Association, Southern Bldg., Washington, D. C.

1. *Building Materials and Supplies.* (manufacturing processes* for cement, ceramics, glass, lumber, etc.)

2. *Chemicals* (manufacturing processes*).

3. *Communications.* Radio tubes and apparatus (manufacturing processes*). Telephone apparatus (manufacturing processes*). Telegraph apparatus (manufacturing processes*). Telephone exchanges (atmospheric control). Broadcasting studios (atmospheric control). Printing and publishing (processing*).

4. *Drugs and Pharmaceuticals* (manufacturing processes*).

5. *Dry Ice* (manufacturing processes*).

6. *Electric Power* (manufacturing processes* for electric motors and other electrical equipment).

7. *Electrical Appliances* (manufacturing processes*).

8. *Explosives* (manufacturing processes*).

9. *Flowers* (growing and sale of).

10. *Furs* (storage).

Foodstuffs

11. *Foodstuffs.* (a) *Processing* of Foodstuffs:* Meat and meat products packing or processing*. Dairy products processing*. Vegetable and fruit processing*. Flour processing*. Baking processes*. Cereals processing*. Ice cream (production and distribution). Sugar. Beverages. Candy processing*. Miscellaneous foods processing*. Quick freezing. (b) *Transportation of Foodstuffs:* Railroad car refrigeration. Truck refrigeration. Ocean and inland ships refrigeration. Precooling. (c) *Distribution of Foodstuffs:* Preservation, storage, and sanitation equipment for 1. Storage warehouses. 2. Wholesale depots. 3. Retail stores. 4. Eating places. 5. Locker storage.

Housing

12. *Housing. Residential:* (a) Homes. (b) Apartments. *Commercial:* (a) Stores. (b) Offices. (c) Hotels. (d) Banks.

A group of key service men from the Atlantic and Pacific fleets of the United States navy launched last month upon a six week course of instruction in the operation and maintenance of air conditioning and refrigerating equipment, conducted by the York Ice Machinery Corp. With air conditioning and refrigeration destined to play an important part in World War II, the navy considers it a vital necessity that its men receive specialized training in this essential field

* * *

(e) Unclassified. *Industrial:* (a) Factories. (b) Laboratories. (c) Unclassified. *Public:* (a) Auditoriums. (b) Public buildings. (c) Unclassified. *Military:* (a) Cantonments. (b) Unclassified. *Institutions:* (a) Educational. (b) Hospitals. (c) Unclassified.

13. *Hospitals and Institutions.* Food and medicine preservation. Control of operating room atmospheric conditions. Control of convalescing space atmospheric conditions.

14. *Ice.* Production. Storage.

15. *Laboratories.*

16. *Leather and Leather Products.*

17. *Malt Beverages.* (Beer, ale, etc.)

18. *Machines and Machine Tools* (production of). Abrasives processing.* Precision and tolerances of parts control.

Metals

19. *Metals.* Copper mining. Zinc mining. Gold mining. Other minerals mining. Iron and steel production (blast furnaces, etc.) Other metals production.

20. *Metal Working*—Ferrous.

21. *Metal Working*—Non-ferrous.

22. *Mortuaries.*

23. *Munitions* (manufacturing processes*).

24. *Optical Apparatus* (processing* of lenses, etc.).

25. *Paper and Paper Products* (manufacturing processes*).

26. *Petroleum Products* (processing*).

27. *Photographic Materials* (manufacturing processing*). Film. Cameras. Other apparatus.

28. *Precision Instruments* (manufacturing processes*).

29. *Recreation.* Theaters. Soda fountains. Bars. Clubs. Skating rinks. Unclassified.

30. *Rubber and Rubber Products* (manufacturing processes*).

31. *Synthetic Products.* Plastics. Synthetic rubber. Unclassified.

Textiles and Clothing

32. *Textiles and Clothing* (manufacturing processes*). (a) *Non-synthetic:* Yarn spinning. Fabric weaving. Garment making. Hosiery making. Parachute making. Balloon making. Other fabric production. (b) *Synthetic:* Base materials manufacture. Yarn spinning. Fabric weaving. Garment making. Parachute making. Other fabric production.

33. *Tobacco Products* (manufacturing processes*).

34. *Transportation* (manufacturing processes*). Motors and engines production. Airplane production. Automobile production. Truck production. Tank production. Railroad car production. Ships and boats production.

*"Processing" denotes operations for which equipment is essential for purposes of controlling chemical, biological, or other reactions of given hygroscopic and non-hygroscopic substances, materials or products. Whenever improved efficiency, health and comfort of workers also results, this is purely an incidental additional advantage.

* * *

Election of Cloud Wampler of Chicago as executive vice-president of Carrier Corp. was announced last month. Mr. Wampler, who is president of Stern, Wampler & Co., Inc., investment banking firm, has been intimately associated with Carrier Corp. since 1932. . . . Henry Knowlton has been appointed sales promotion manager of the Airtemp Div., Chrysler Corp. He will study the application of air conditioning, refrigeration and heating equipment to selected vocational fields as part of the company's new dealer development and field training program. . . . Paul B. Zimmerman has been appointed vice-president and general sales manager of Airtemp. . . . Edmund R. Walker has been promoted to assistant general manager of the Fedders Mfg. Co. He was formerly manager of the air conditioning division. . . . C. W. Pearsall has been appointed general sales manager of Ahlberg Bearing Co. . . . Herbert F. Curtis was recently appointed sales manager of the Auer Register Co.

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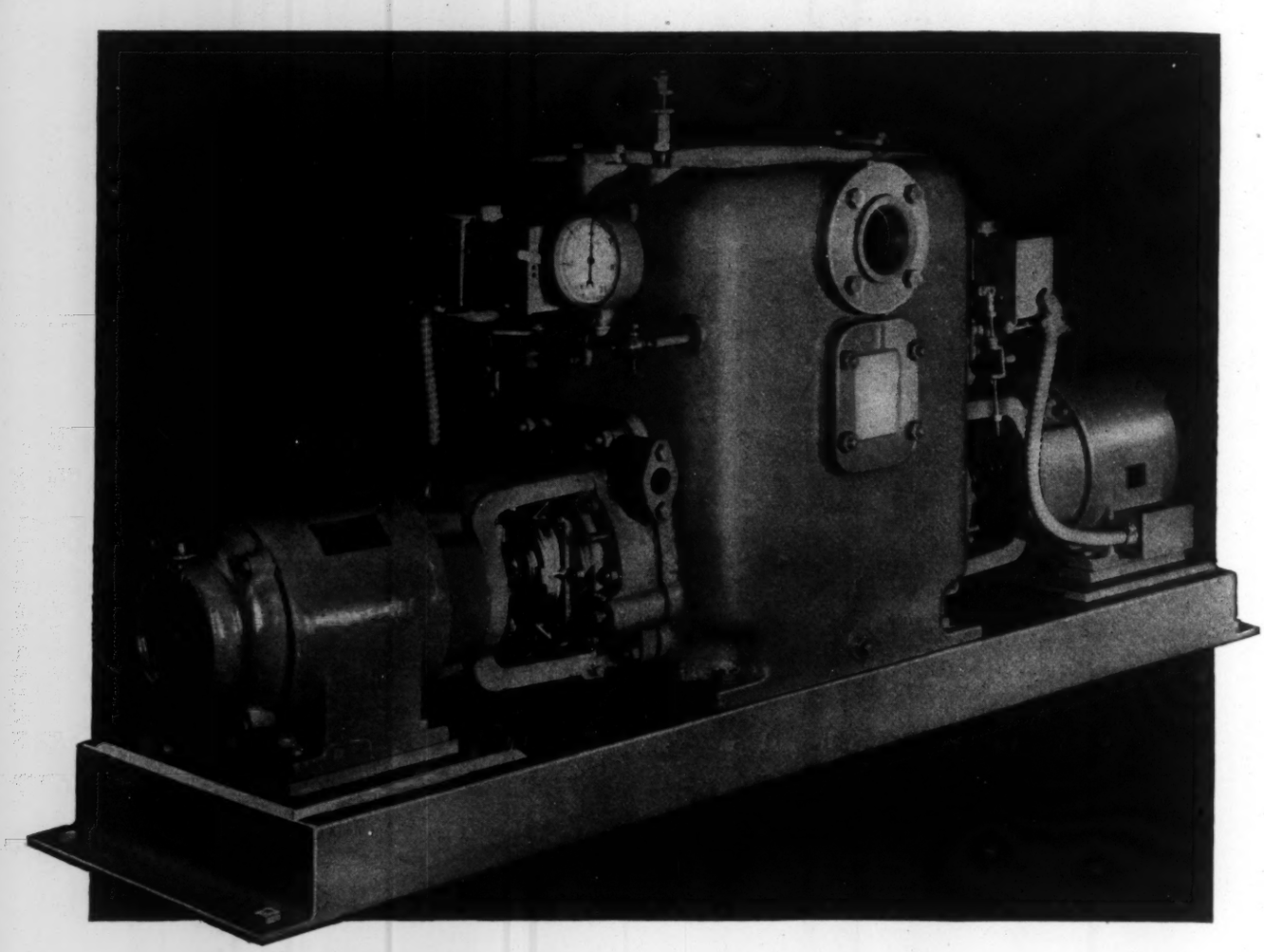

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SOCKET WELDING FITTINGS STANDARDIZED

Sabin Crocker and G. A. Daeuble, Jr.** Explain Highlights of the New Proposed American Standard*

EARLY in the development of welded piping systems the need was sensed for a line of socket welding fittings for use with small size pipe. The particular advantage of socket type fillet welds over butt welds for this application is that complete fusion between socket and pipe is readily obtainable without the use of backing rings which tend to obstruct unduly the bore of small diameter pipe. Another advantage of a socket weld is that the pipe end does not require beveling but slips into and is supported by the socket, thus becoming self aligning and obviating the necessity of tack welding or clamping the parts prior to welding. Furthermore, where streamline flow is not absolutely essential, the depth of socket bore provides some come and go so that it is unnecessary to cut the pipe to accurate length, or even to cut it off square. Hence the growing interest in the development of socket welding fittings has expanded to keep pace with the rapid adoption of welded joints for all service conditions.

Need for Standardization

It frequently happens during the early development of a new product that different designs and dimensions are offered for equivalent articles turned out by competing manufacturers or specified by independent designers. As experience is obtained with the product and opinion crystallizes on the relative merits of conflicting features, the trade eventually demands that design be boiled down to some generally acceptable compromise and basic dimensions standardized to

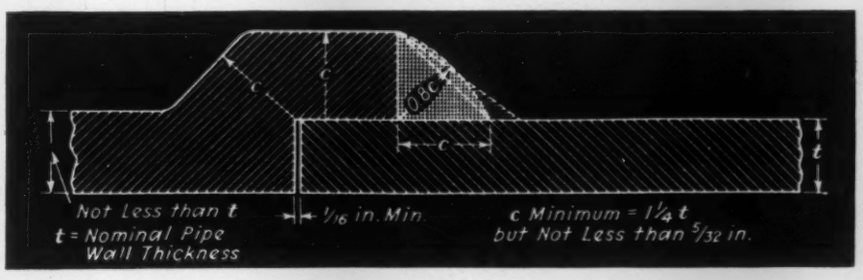
SOCKET FITTINGS — One thing for which the piping industry should be particularly noted is the outstanding job of standardization it has done in recent years. Those who have followed the pages of HEATING, PIPING AND AIR CONDITIONING regularly have read many articles giving the background and highlights of the various new standards . . . . . Many of these articles have been written by Mr. Crocker, an authority who is most active in piping standardization work. We are pleased to present this month a discussion by Mr. Crocker and Mr. Daeuble of the latest piping standardization project — the proposed American Standard for Socket Welding Fittings. They describe its provisions and background in some detail, and a figure and three tables from the standard are reproduced. As noted at the end of their article, copies of the standard itself may be had by those who wish to comment on it, and the committee responsible for its development will welcome such criticism and comment . . . . . Sponsors of the new socket fittings standard are the Heating, Piping and Air Conditioning Contractors National Association, the Manufacturers Standardization Society of the Valve and Fittings Industry, and the American Society of Mechanical Engineers

suit. So it was in the case of socket welding fittings, where conflicting schools of thought argued the merits of deep *vs.* shallow sockets — thus introducing a variety of overall dimensions which precluded interchangeable use of different brands of fittings. This situation was bad for producers and consumers alike in that it tended to obstruct free competition through practically requiring each consumer either to settle on one brand of fitting and use it to the exclusion of all others or else to set up his own dimensional standard for the product.

Socket welding fittings first appeared on the market as a commercial article about 1934, although fittings of this sort had been tried out by the U. S. Navy at least as early as 1928. By 1938 confusion arising from the variety of dimensions in which the product was supplied had reached proportions which led to a request for its standardization being entertained by ASA sectional committee B16. In view of the generally recognized need for establishing standard dimensions, the project was referred by the sectional committee to its subcommittee No. 3 which organized a subgroup on welding fittings and called a meeting of those interested in December, 1938. Owing, however, to discordant dimensions which necessitated changes in manufacturing equipment by several manufacturers before an interchangeable product could be offered, some time has been required to effect satisfactory compromises and formulate a proposed standard for submission to sectional committee B16 and industry for acceptance.

As far as is known, the manufacturers of socket welding fittings have been 100 per cent represented on the working committee which formulated this dimensional standard and they have been unanimous in agreeing to conform. In some

Fig. 1—Minimum requirements for socket and fillet weld dimensions



*Senior engineer, engineering div., The Detroit Edison Co. Member of HPAC's board of consulting and contributing editors. Chairman of subgroup No. 6 on standardizing the dimensions of welding fittings, to which subgroup ASA sectional committee B16 has assigned this job.

**Vice-president, Henry Vogt Machine Co. Chairman sub-subgroup No. 2 on socket welding fittings of subgroup No. 6.

cases this has entailed a considerable sacrifice in scrapping existing tool equipment or in superseding or relegating to a special classification certain lines of nationally advertised fittings. The manufacturers of this product are to be commended for the coöperative spirit of give and take manifest by them throughout the whole affair. The standard already has been put into effect by several of the manufacturers of socket welding fittings who have changed their equipment to conform and are now supplying their product to the proposed new dimensions. It is hoped that the piping trade in general will find the proposed standard basically acceptable without alteration beyond some possible minor revisions of dimensions.

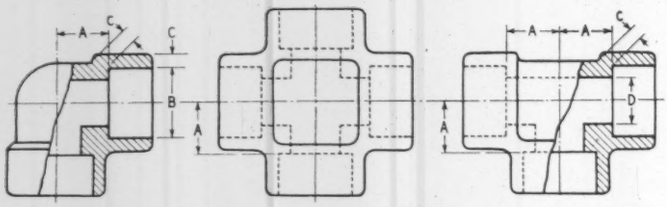


Table 1 Dimensions of Socket Welding Elbows, Tees, and Crosses

Nominal Pipe Size	Center to Bottom of Socket for Ells, Tees, Crosses	Bore Diameter of Socket	Socket Wall Thickness			Bore Diameter of Fitting		
			Sched 40	Sched 80	Sched 160	Sched 40	Sched 80	Sched 160
	A	B	C			D		
3/8	*17/32	*0.690	0.156	0.158		0.493	0.423	
1/2	5/8	0.855	0.156	0.184	0.234	0.622	0.546	0.466
3/4	3/4	1.065	0.156	0.193	0.273	0.824	0.742	0.614
1	7/8	1.330	0.166	0.224	0.313	1.049	0.957	0.815
1 1/4	1 1/16	1.675	0.175	0.239	0.313	1.380	1.278	1.160
1 1/2	1 1/4	1.915	0.181	0.250	0.351	1.610	1.500	1.338
2	1 1/2	2.406	0.193	0.273	0.429	2.067	1.939	1.689
2 1/2	1 5/8	2.906	0.254	0.345	0.469	2.469	2.323	2.125
3	2 1/4	3.535	0.270	0.375	0.546	3.068	2.900	2.626

All dimensions are given in inches.
Dimension "C" is 1¼ times the nominal pipe thickness, minimum.
* This dimension applies to Schedules 40 and 80 only.

Fillet Weld Stronger Than Pipe

While the chief interest in socket welded joints centers in small pipe below 3 in. nominal size, this type of joint has been used successfully in the OD pipe sizes and even for large diameter steel pipe for water and gas service, where such joints are termed "bell and spigot." According to a published report made in 1928 to the American Gas Association by its subcommittee on pipe joints:

The simple butt welded joint is the most efficient (for joining parts of equal thickness) and is recommended for use in welding (large) pipe. . . . . Very high strength and nearly equally satisfactory results were obtained with socket . . . . joints (in 4, 8, 12 and 16 in. OD pipe). [*The parenthetical explanation was added by the writer as being readily apparent from the context.*]

With reference to the foregoing quotation and parenthetical observation about "joining parts of equal thickness" it is significant to call attention to Fig. 1 of the proposed standard wherein it is provided that the thickness of the socket *C* shall be at least 1¼ times the pipe wall thickness and that the throat of the weld shall be at least 0.8 *C*, which makes the throat equal to the pipe wall thickness. The intent of this provision is to insure having the fillet weld as strong as the pipe. Some extra margin is available in small size pipe where the conoidal area through the throat of the weld is considerably greater than the annular section through the wall of the pipe owing to the greater circumference of the former. As an added precaution in obtaining strong joints in Schedule 40 pipe smaller than 1 in. nominal size, it is required that the thickness of socket shall in no case be less than 5/32 in.

Numerous tests have demonstrated that the strength of socket welds having an area of throat at least equal to the area of pipe wall surpasses that of the pipe both under bursting pressure, and under combined tensile and shear tests†. Hence the design of fillet weld covered in the standard is deemed amply strong, particularly so within the range of ⅜ in. to 3 in. nominal pipe sizes to which the standard is now confined.

It is of interest to note that the proposed standard for socket welding fittings meets the following requirements quoted from the 1940 issue of the ASME Boiler Construction Code.

Socket type joint connections of pipes to valves or fittings may be used provided the nominal pipe size does not exceed 4 in.; the depth of the socket is not less than ¼ in.; the end of the pipe before welding is not less than 1/16 in. from the bottom of the socket; the throat dimension of the fillet weld is not less than the thickness of the pipe wall; the hub thickness is not less than 1¼ times the nominal pipe thickness; and the weld is not in contact with the furnace gases.

The proposed new draft of the American Standard Code for Pressure Piping (ASA-B31.1—) now under consideration sanctions fillet welds in the following language:

(Section 624c) Fittings . . . . may be attached by fillet welds, provided the pipe is inserted in the fitting and provided the fillet welds meet the requirements of Fig. 11. [*Fig. 11 is the same as Fig. 1 reproduced here in connection with the Proposed American Standard for Socket Welding Fittings.*]

Basic Dimension

Center to Bottom of Socket

Owing to conflicting opinions among manufacturers concerning depth of socket, some difficulty was encountered at the outset in settling on a generally acceptable basis for overall dimensions. The following possibilities were considered: (a) Centerline of fitting to bottom of socket. (b) Centerline of fitting to end of pipe used with fitting. (c) Centerline of fitting to outboard end of socket.

After much discussion possibility (a) was settled on as the most logical since, from the standpoint of the user, this dimension governs the location of the fitting with reference to the entire piping layout. Difficulty in arriving at a standard dimension for center to bottom of

†For test results see: (a) Application of Welding for Piping and Pipe Fittings, bureau of construction and repair, U. S. Navy technical bulletin No. 7-29, October, 1929. (b) Investigation of Gas and Arc Fillet Welds in Piping, by Irving H. Carlson and Eric R. Seabloom, American Welding Society paper, October, 1940, published in The Welding Engineer, November, 1940, pp. 846-854.

socket due to difference in existing manufacturing practice was the occasion for considering possibilities (b) and (c). Neither of those alternates was deemed a satisfactory substitute, however, since gaging and inspection would be awkward for (b), while (c) failed to establish for the user a length of pipe suiting different brands of fittings having varying depths of socket.

Hence it was finally agreed, at the expense of having to discard some manufacturing equipment, to standardize on the centerline of fitting to bottom of socket as the basic dimension and let the end of the socket extend over the pipe by varying amounts corresponding to the depth of socket desired by the individual manufacturers. Although the depth of socket, within reasonable limits, was considered unimportant by most users, considerable manufacturing equipment and some sales effort had been devoted by certain manufacturers towards furthering shallow or deep sockets according to their individual preferences.

Adoption of the centerline of fitting to bottom of socket principle was coupled with a proviso that the sockets must be at least as deep as the minimum dimension prescribed in section 7(b) of the proposed standard. This left the way open to accommodate the preferences of those manufacturers who advocated some particular depth of socket, and at the same time made all brands of fittings interchangeable from the users' standpoint. Incidentally, for any given pipe size, all weights of fittings (*viz.*, Schedules 40, 80 and 160) have the same dimensions for diameter of socket bore, for center of fitting to bottom of socket (except for 45 deg elbows), and for minimum depth of socket. As might be expected this leaves only the wall thicknesses of fitting and socket, and the bore diameter of the fitting proper to vary with the schedule number.

The foregoing features are illustrated in Fig. 2, a photograph of four 1½ in. fittings (reduced to about two-thirds actual size) which, after welding stub ends of pipe into one socket of each fitting, have been sawed in two longitudinally and etched. Fittings *A* and *B* both are Schedule 40 but of different makes, thus serving to show permissible differences in depth of socket and wall thickness which are contemplated within the scope of the standard. Schedule 40, 80 and 160 fittings of a single make are illustrated in *B*, *C* and *D*, respectively. Here can be observed identical center line to bottom of socket dimensions, and identical socket bores, but progressively smaller bores in the fitting proper. Incidentally, for Schedule 160 fittings, this manufacturer uses sockets considerably deeper than the minimum dimension specified alike for all weights embraced in the standard. This is apparent in comparing fitting *D* with *B* and *C*. Likewise, fitting *A* of a different make has a slightly deeper socket than *B* and *C*.

Attention is directed also to the attempt made in preparing the welded specimens to maintain the required 1/16 in. spacing between the pipe end and bottom of socket, and to get the full 0.8*C* throat dimension across the weld as shown in Fig. 1 of the proposed standard. [*Fig. 1 of the proposed standard is reproduced here also as Fig. 1.*]

Service Ratings

It is intended that socket welding fittings shall be as strong as the corresponding pipe of the designated schedule number. (Table 3 reproduced here will appear in the appendix to the standard.) The wall thickness of fittings must be equal to or greater than the wall thickness of the designated pipe schedule. Rules for establishing the strength, and indirectly the metal thickness, of fittings through test of pilot fittings are set forth in detail in section 6 of the proposed standard. Pressure ratings of welding fittings can be determined only indirectly through use of the *S* values for pipe under various service conditions as established in the American Standard Code for Pressure Piping and the ASME Boiler Construction Code. This is in keeping with present day ideas on suiting ratings to the great variety of service conditions and materials now encountered. By way of identifying these new fittings with familiar old pipe designations, it is of interest to recall that Schedule 40 agrees approximately with standard weight pipe, and Schedule 80 with extra strong.

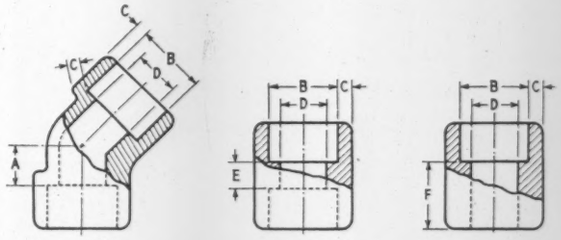


Table 2 Dimensions of Socket Welding 45 Deg Elbows, Couplings, and Half Couplings

Nominal Pipe Size	Center to Bottom of Socket for 45 Deg Ells		Couplings, Distance Between Bottoms of Sockets	Half Couplings, Bottom of Socket to Opposite Face	Bore Diameter of Socket	Socket Wall Thickness			Bore Diameter of Fitting		
	Sched 40 and 80	Sched 160				Sched 40	Sched 80	Sched 160	Sched 40	Sched 80	Sched 160
	A		E	F	B	C			D		
⅜	5/16	.....	*¼	*11/16	*0.690	0.156	0.158	.....	0.493	0.423	.....
½	⅜	7/16	⅜	⅞	0.855	0.156	0.184	0.234	0.622	0.546	0.466
¾	7/16	½	⅜	15/16	1.065	0.156	0.193	0.273	0.824	0.742	0.614
1	½	⅝	½	1⅛	1.330	0.166	0.224	0.313	1.049	0.957	0.815
1¼	⅝	¾	½	1 3/16	1.675	0.175	0.239	0.313	1.380	1.278	1.160
1½	¾	13/16	½	1¼	1.915	0.181	0.250	0.351	1.610	1.500	1.338
2	13/16	1⅛	¾	1 11/16	2.406	0.193	0.273	0.429	2.067	1.939	1.689
2½	1⅛	1¼	¾	1 11/16	2.906	0.254	0.345	0.469	2.469	2.323	2.125
3	1¼	1¼	¾	1¾	3.535	0.270	0.375	0.546	3.068	2.900	2.626

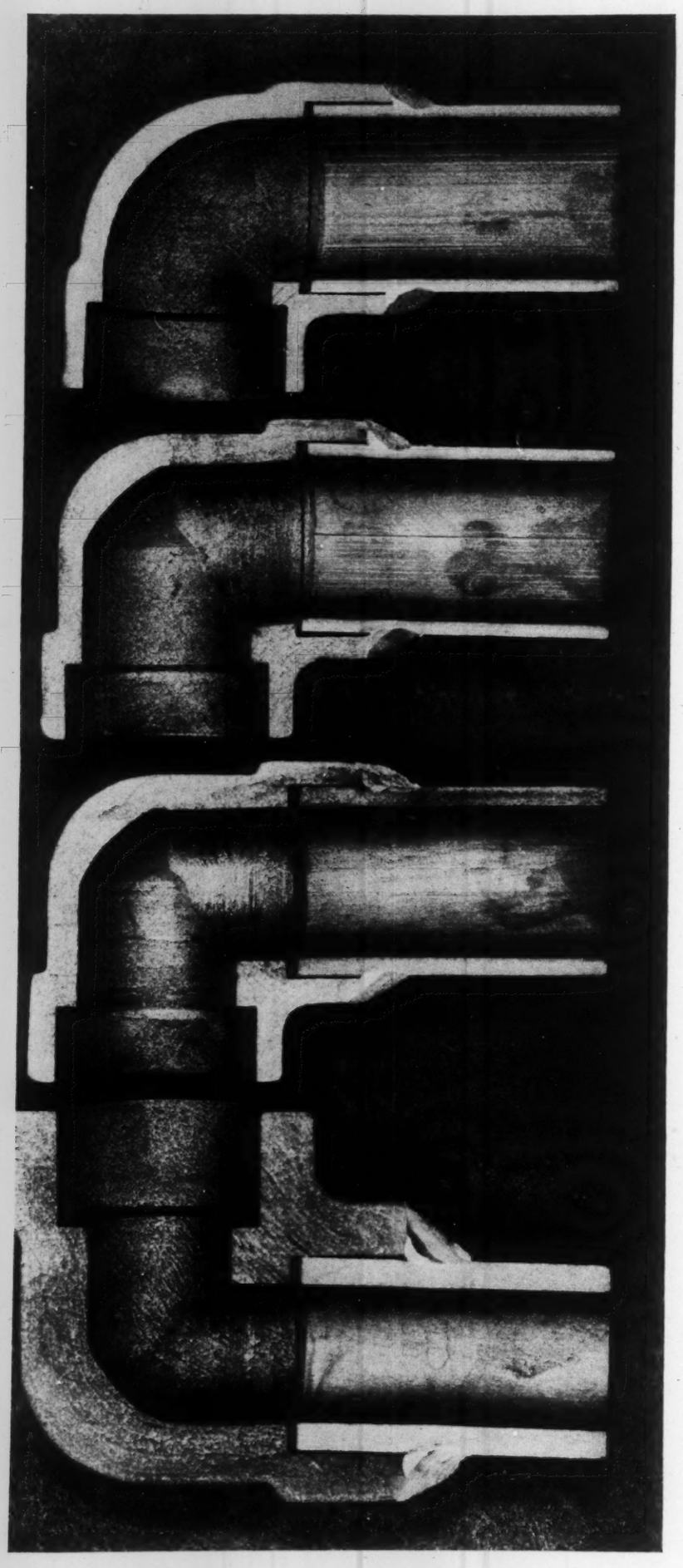
All dimensions are given in inches.

Dimension "C" is 1¼ times the nominal pipe thickness, minimum.

* This dimension applies to Schedules 40 and 80 only.

Material Specifications

According to section 5 of the proposed standard, the material of wrought fittings shall be in accordance with ASTM specification A234, and of cast fittings in accordance with ASTM specification A216 for welding grade carbon cast steel or A217 for welding grade alloy



cast steel. In this standard "wrought" is used to denote fittings made of pipe, tubing or forgings.

Specification A234 for Factory Made Wrought Carbon Steel and Carbon Molybdenum Steel Welding Fittings was first issued by the ASTM as a tentative specification in 1940. Formulation of this specification was begun by the ASTM in 1937 as a companion job to the ASA dimensional standard for butt welding fittings which was started about the same time. Later the scope of the materials specification was extended to embrace socket welding fittings as well. The story of ASTM specification A234 by Chairman J. J. Kanter of the subcommittee which formulated it appeared in HEATING, PIPING AND AIR CONDITIONING for April and May, 1941.

Hydrostatic Testing

For pressure containing parts, hydrostatic testing often is considered more significant than tensile testing of the material itself; this is particularly pertinent where a non-destructive test is desired in order to obviate having to cut up the finished article in obtaining tensile test specimens. The materials specifications already referred to contemplate that only material of a given quality whose properties are determined by chemical analysis and tensile testing shall be used in the manufacture of factory made welding fittings. Further tensile testing of the finished welding fitting is deemed superfluous. Routine hydrostatic testing of wrought welding fittings is not required by the proposed standard, although on special agreement they can be tested according to rules laid down in section 8. All cast welding fittings must be tested as provided in ASTM specification A216 or A217.

As stated in section 6 of the proposed standard, however, the actual bursting strength of each manufacturer's line of fittings must be determined in advance of going on the market through hydrostatic tests on

Fig. 2 — Proposed American Standard socket welding fittings, 1½ in. nominal pipe size, etched sections about two-thirds actual size. Fitting *A*, Schedule 40, Manufacturer 1. Fitting *B*, Schedule 40, Manufacturer 2; Fitting *C*, Schedule 80, Manufacturer 2; Fitting *D*, Schedule 160, Manufacturer 2

pilot fittings of each size, schedule and type. To insure adequacy of fitting design, the actual bursting strength of fittings must be demonstrated to equal the computed bursting strength of pipe of the designated schedule number and material.

Standard Fitting Types

As indicated in connection with Tables 1 and 2 of the proposed standard, which are reproduced here, the types of socket welding fittings so far standardized are 90 deg elbows, 45 deg elbows, tees, crosses, couplings and half couplings. Reducing sizes are contemplated and shall have the same centerline of fitting to bottom of socket dimension as the largest size of the reducing fitting. The members of subgroup No. 6 are aware of a considerable interest in having standardized the center to bottom of socket dimension of laterals and valves having socket welding ends. It was the consensus, however, that standardization of these items would have to await on issuance of the proposed standard and development of a greater public demand for them.

Other Items Within Scope of Proposed Standard

The scope of this standard is stated to cover overall dimensions, tolerances and marking for socket welding fittings. Actually, the full scope embraces finish, minimum requirements for socket and fillet weld dimensions, and testing as well. Most items within this full scope already have been discussed in this article or are explained in full detail in the proposed standard. Beyond this, some mention should be made of the fact that socket welding fittings are intended to match the commercial accuracy in dimension of the pipe with which they are used. Hence, considerable thought has been given to specifying pertinent dimensions, to calling for finish where required for accuracy of fit or facility of welding, and to establishing reasonable tolerances on all dimensions.

Table 3

Dimensions of Welded and Seamless Steel Pipe (ASA B36.10–1939)

Nominal Pipe Size	Outside Diameter	Nominal Wall Thicknesses for Schedule Numbers									
		Sched 10	Sched 20	Sched 30	Sched 40	Sched 60	Sched 80	Sched 100	Sched 120	Sched 140	Sched 160
⅛	0.405	...	...	...	**0.068**	...	**0.095**	...	...	...	...
¼	0.540	...	...	...	**0.088**	...	**0.119**	...	...	...	...
⅜	0.675	...	...	...	**0.091**	...	**0.126**	...	...	...	...
½	0.840	...	...	...	**0.109**	...	**0.147**	...	...	...	0.187
¾	1.050	...	...	...	**0.113**	...	**0.154**	...	...	...	0.218
1	1.315	...	...	...	**0.133**	...	**0.179**	...	...	...	0.250
1 ¼	1.660	...	...	...	**0.140**	...	**0.191**	...	...	...	0.250
1 ½	1.900	...	...	...	**0.145**	...	**0.200**	...	...	...	0.281
2	2.375	...	...	...	**0.154**	...	**0.218**	...	...	...	0.343
2 ½	2.875	...	...	...	**0.203**	...	**0.276**	...	...	...	0.375
3	3.5	...	...	...	**0.216**	...	**0.300**	...	...	...	0.437
3 ½	4.0	...	...	...	**0.226**	...	**0.318**	...	...	...	...
4	4.5	...	...	...	**0.237**	...	**0.337**	...	0.437	...	0.531
5	5.563	...	...	...	**0.258**	...	**0.375**	...	0.500	...	0.625
6	6.625	...	...	...	**0.280**	...	**0.432**	...	0.562	...	0.718
8	8.625	...	0.250	**0.277**	**0.322**	0.406	**0.500**	0.593	0.718	0.812	0.906
10	10.75	...	0.250	**0.307**	**0.365**	**0.500**	0.593	0.718	0.843	1.000	1.125
12	12.75	...	0.250	**0.330**	0.406[1]	0.562[1]	0.687	0.843	1.000	1.125	1.312
14 OD	14.0	0.250	0.312	0.375	0.437	0.593	0.750	0.937	1.062	1.250	1.406
16 OD	16.0	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.437	1.562
18 OD	18.0	0.250	0.312	0.437	0.562	0.718	0.937	1.156	1.343	1.562	1.750
20 OD	20.0	0.250	0.375	0.500	0.593	0.812	1.031	1.250	1.500	1.750	1.937
24 OD	24.0	0.250	0.375	0.562	0.687	0.937	1.218	1.500	1.750	2.062	2.312
30 OD	30.0	0.312	0.500	0.625	...	...	...	...	...	...	...

All dimensions given in inches.

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions. For tolerances on wall thicknesses see appropriate material specifications.

Thicknesses shown in bold face type in Schedules 30 and 40 are identical with thicknesses for "standard weight" pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for "extra strong" pipe in former lists.

Reprinted from American Standard for Wrought-Iron and Wrought-Steel Pipe, ASA B36.10-1939.

[1]Owing to a necessary departure from the old "standard weight" and "extra strong" thicknesses in these two sizes, the new thicknesses are not as yet stocked by all manufacturers and jobbers. Hence, where agreeable to the purchaser and suitable for the service conditions, the old "standard weight" 0.375 in. wall pipe corresponding to a 1000 P/S value of 37.7 is still available and can be substituted for the 0.406 in. wall, and the old "extra strong" 0.500 in. wall pipe corresponding to a 1000 P/S value of 55 can be substituted for the 0.562 in. wall.

Acceptance by Industry

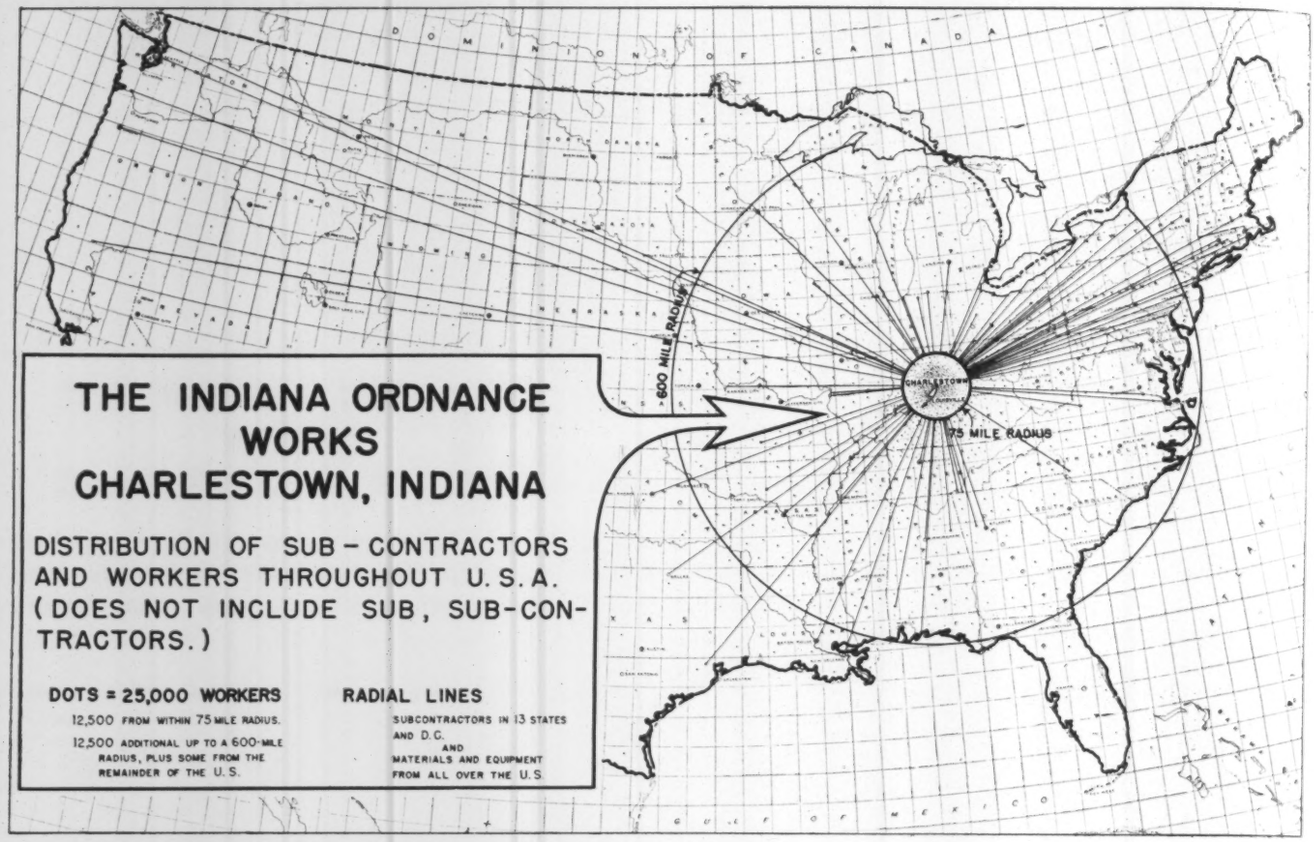
Following unanimous approval by the membership of subgroup No. 6, page proof copies of a tentative draft of the Proposed American Standard for Socket Welding Fittings (ASA B16.11-) have been submitted to all members of ASA sectional committee B16 and to about 100 firms representative of industry or concerned with the manufacture or use of the product. If criticism for improving the standard is forthcoming, it will be considered by the various committees concerned and incorporated where practicable in a revised draft. Should anyone reading this article wish to comment on any requirements of the tentative draft, his ideas will be welcome if addressed to Subgroup No. 6 on Welding Fittings in care of the American Society of Mechanical Engineers, 29 W. 39th St., New York, N. Y. As noted on p. 419 of the July HPAC, copies of the proposed standard may be obtained from the ASME.

HEAT FLOW THROUGH SUNLIT WALLS

Parts 4 and 5 of a series of articles by Professor C. O. Mackey and L. T. Wright, Jr. of the Sibley school of mechanical engineering at Cornell university on Estimating Heat Flow through Sunlit Walls have been issued as reprint No. 14 by the Engineering Experiment Station.

The first three articles, collected in a previous reprint, set forth a rational method of determining the contribution to the cooling load of solar radiation and air temperature differences at different hours of the day, as applied to walls facing north, east, south and west. The two collected in the new reprint present graphs and tables for walls facing northeast, southeast, southwest and northwest, and also compare the results of the Mackey and Wright method with those obtained from methods previously employed.

Materials considered are brick, masonry, concrete, and wood frame with various types of insulation. Reprints are available without charge on request to the Engineering Experiment Station, Cornell University, Ithaca, N. Y.


THE INDIANA ORDNANCE WORKS
CHARLESTOWN, INDIANA
DISTRIBUTION OF SUB-CONTRACTORS AND WORKERS THROUGHOUT U. S. A. (DOES NOT INCLUDE SUB, SUB-CONTRACTORS.)
DOTS = 25,000 WORKERS
12,500 FROM WITHIN 75 MILE RADIUS.
12,500 ADDITIONAL UP TO A 600-MILE RADIUS, PLUS SOME FROM THE REMAINDER OF THE U. S.
RADIAL LINES
SUBCONTRACTORS IN 13 STATES AND D.C.
AND
MATERIALS AND EQUIPMENT FROM ALL OVER THE U. S.
600 MILE RADIUS
75 MILE RADIUS
CHARLESTOWN
DOMINION OF CANADA
GULF OF MEXICO

Sixty-five subcontractors, themselves requiring services of hundreds of sub-subcontractors, are represented in the smokeless powder plant at Charlestown, Ind., 15 miles north of Louisville, Ky. This chart shows points in the United States which have supplied material and workers for the project. A recent visitor saw cars from 36 states in the parking lot

ARSENAL OF DEMOCRACY

"War is no longer Samson with his shield and spear and sword, and David with his sling; it is no longer selected parties representing nations as champions, and in physical conflict one with the other; it is the conflict of smokestacks now, of the driving wheel and the engine."—Newton D. Baker, quoted by Undersecretary of War Robert P. Patterson in a recent address.

THE first two of an ultimate six production lines at the Charlestown, Ind., Ordnance Works recently went into production for the "conflict of the smokestacks." When complete, this "arsenal of democracy" will be the world's largest smokeless powder plant and will have capacity to manufacture powder in sufficient quantity to supply the needs of an army of over 1,000,000 men. Smokeless powder is known as a "propellant", and furnishes the force to propel a shell from a gun—as contrasted with the explosive in the shell, which does its work at the shell's destination. Briefly—very briefly—smokeless powder is nitrocellulose obtained by the addition of nitric acid to cellulose.

The Charlestown plant is of permanent construction, and is designed to supply the needs of our army—even after the present emergency—for this essential but "noncommercial" item of ordnance. No longer need smokeless powder be considered the most serious bottleneck in America's defense program. Built by E. I. du Pont de Nemours & Co. for the army, in coöperation with a skilled group of army ordnance department engineers and experts under the direction of Lieutenant-colonel R. E. Hardy, it will be operated for the army by the du Pont company on a fixed fee basis.

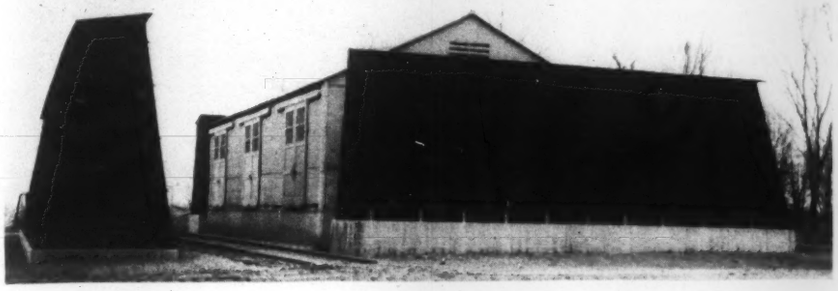
Site of the huge plant, which comprises some 600 buildings, is a 5500 acre tract about 15 miles from Louisville, Ky., which — until

—for the Conflict of Smokestacks

WORLD'S LARGEST POWDER PLANT WILL SUPPLY NEEDS OF AN ARMY OF OVER 1,000,000 MEN

ARSENAL — When completed, the Charlestown, Ind., Ordnance Works — near Louisville, Ky.—will be the world's largest smokeless powder plant. Of permanent construction, it will have capacity to supply the needs of an army of over a million men for this essential "non-commercial" item of ordnance . . . The power and process and heating steam requirements of this huge manufacturing establishment are furnished by two large power plants fired by pulverized coal. Many miles of piping — steam, water, hydraulic, refrigeration, process, etc. — are required. Heating of the manufacturing buildings is generally by unit heaters. Air conditioning is essential in the ballistics laboratory, where closed bomb tests are run, in order to assure accuracy of delicate measuring operations. There are extensive air handling systems for drying, and for recovering solvent, at several points in the process

ground was broken last September —was corn field and tobacco patch. Total cost of the plant will be approximately $90,000,000 when completed, and 9000 employees will be required for full operation. In addition to the six big production lines and the magazines, the major buildings of the plant include a hospital, cafeteria, sewage treatment plant, telephone exchange, administration building, guard staff garage, fire house, ballistics laboratory, ordnance inspectors' laboratory and various field laboratories.



Shipping house, with barricade to isolate effects of a possible explosion

The site was chosen to be far enough inland to be relatively safe from bombing attack from the air, near enough to draw upon the industrial populations of Louisville and nearby Indiana cities, and far enough from metropolitan areas to present no hazards to civilians. The little town of Charlestown has become a relatively small collection of buildings across the highway from the plant's main entrance—a village whose backyards have been packed hub to hub with trailers used for housing hundreds of the construction workers. Other thousands of workers come to the job each day by work trains, automobile, bus and truck.

The plant site also has the advantages of good rail and road transportation, accessibility to raw materials, and the Ohio River for water supply if needed, and waste disposal. Cotton linters can be transported by river from the South, a new government ammonia plant is also being built near Louisville, and large sulphuric acid plants are situated in Tennessee. Alcohol and ether are also available in the Midwest.

The output of the Charlestown plant will go to a bag loading plant being built on an adjacent tract, and to shell loading plants in Indiana, Ohio, Illinois and Iowa.

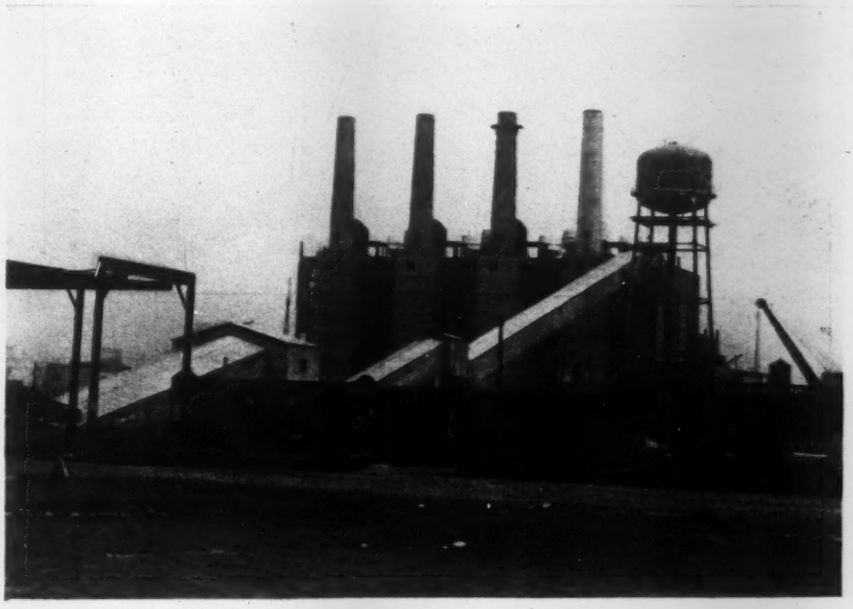
Power and Steam

The power and process and heating steam requirements of the ordnance works are to be supplied by two large power plants—one with five 160,000 lb per hr units and the other with six such units—generating steam at 550 psi, and fired by pulverized coal.

All water for the power plants is treated by the zeolite process and is degasified. Continuous blowdown is used on the boilers. Make-up is 95 per cent.

The water requirements of the ordnance works — approximately three times those of the nearby city of Louisville—are to be furnished by seven collectors of the radial well type. The water (63,000,000 gal per day) is carried 2¾ miles to the plant by several 36 in. cast iron cement lined mains, where it is stored in two 5,000,000 gal reservoirs and in three special 150,000 gal reser-

One of the two power plants serving the Charlestown Ordnance Works, shortly before completion



voirs—one for fire protection, and the other two for the power plants. The water is of uniform temperature the year around, and has a low iron content.

Piping Services

Many miles of piping are required throughout the plant. Steam at 150 and 50 psi is distributed for process and heating from the power houses in gas or arc welded piping, both welding methods being used. Brine is piped from a central ammonia refrigeration plant for cooling jackets on autoclaves. There are 3500 psi hydraulic power lines, air lines, water lines and process piping of stainless steel, lead and other metals for various materials—which include caustic, ether, alcohol, acids, recovered solvent, etc.

Much of the larger size piping (12 in. and over) was prefabricated in the East and shipped to Charlestown.

Heating

Heating of the manufacturing buildings is generally by means of suspended unit heaters supplied by 50 psi steam from the power plants which is reduced to 30 psi or less.

In the buildings in the administrative area, convectors under thermostatic control or direct radiation are used for heating. One of the buildings in this area uses steam at 2 psi pressure and the others use 30 psi steam, supplied from the power plant.

The cafeteria is heated by unit ventilators.

None of the condensate is returned to the boilers.

Air Conditioning

There is little air conditioning equipment at this powder plant, but an installation in the ballistics laboratory is of unusual interest and is essential to the studies carried on there.

It is of course absolutely essential to have complete and accurate information about the characteristics of the smokeless powder produced. In this laboratory, the closed bomb test is used to determine what the powder will and will not do; a small sample of the powder is burned in a closed vessel and the pressures so generated are measured. Thus, the performance and characteristics of the powder can be predicted.

The pressures are measured electrically and it is for this reason that air conditioning is required. It is necessary—for accurate measurement—to maintain momentarily a static condition in the amplifier circuit. As humidity variations would affect the electrical circuit, air conditioning equipment is installed to maintain a relative humidity "not to exceed 70 per cent" at 75 F dry bulb temperature.

The ballistics laboratory also has a firing range where small ammunition is tested by firing from guns. This firing range—a long narrow room—is heated by pipe coil radiation on the two long walls, which is under thermostatic control in order that fairly constant temperature of the air can be maintained. Considerable variations in temperature of the air in the firing range might affect the accuracy of the test results.

The firing range has double interlocked panels over the apertures through which the bullets pass, for the sake of safety. The exterior door at the target end is also interlocked with these panels so that no one can step into the line of fire from the outside. The valves and the thermostat on the side walls are protected from wild bullets by small projecting plates of steel.

Welded Duct Work

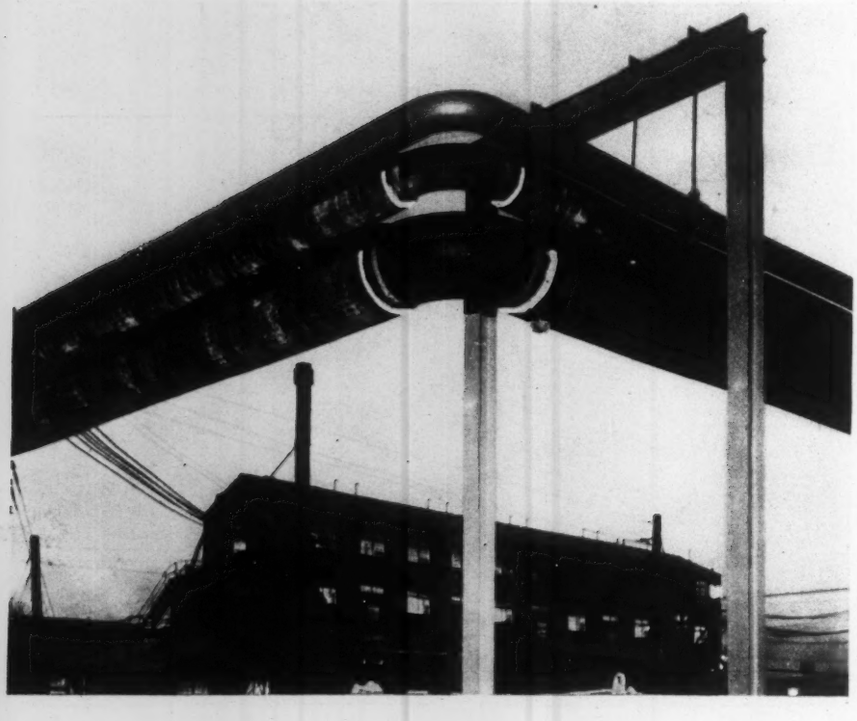
No attempt is made in this discussion to describe details of the manufacturing processes at the Charlestown works—many of them would of course come under the head of military or commercial secrets. For the same reason, no photographs inside any of the buildings could be taken.

As the powder approaches its final form, an important step in the process is the removal of a considerable amount of ether-alcohol solvent. The powder is placed in drying chambers while warm air heated by blast coils is forced through the product in a continuous cycle, the solvent being condensed when the air stream is passed over cooling coils. The air is then reheated.

Special duct systems of this kind comprise several million pounds of sheet metal. The ducts must be as air tight as possible and the connections are therefore welded. Much of this duct work was prefabricated in an eastern plant and shipped to Charlestown in this form, in order to speed up the work.

Final drying operations for removal of the last traces of solvent involve extensive installations of fans, duct work and heating coils using all outside air.

Some of the piping services during installation



Conservation for National Defense

FUEL ECONOMY IN OIL FIRED PLANTS

Kalman Steiner Reviews Methods of Economizing in Generation of Steam

THE President of the United States has proclaimed an unlimited emergency, and has called upon all citizens to assist in a mighty defense movement that can be successful only if it receives the unceasing coöperation of each of us. Chief among the immediate needs of the nation is a concerted effort to direct our vast resources toward vital defense production, so that our armed forces may be equipped with the tools they require to become invincible. Outstanding among the materials essential to industrial production is fuel. Except for a relatively small amount of water power, we are completely dependent upon fuel to keep our factories working and our buildings warm. It is most important that those charged with the use of this prime material exert every possible thought and effort toward conserving it. Operating engineers, obviously, can do more than any other group to reduce the consumption of fuel to a minimum.

The subject of fuel economy is not new to the operating engineer. Most engineers have familiarized themselves with what is considered good practice and have tried to keep abreast of mechanical developments and engineering progress. The literature dealing with the subject is extensive. A short article could not begin to cover even the fundamentals in a detailed way. Instead, this article will present a review of the basic concepts, and point out the directions in which effort can be expended with profit to bring about significant fuel savings. While much of the information given here could be applied to all fuels, it is intended primarily for oil fired plants.

In general, two broad classifications can be made: economies in steam generation and economies in steam utilization (or the equivalent in hot water heating systems). In other words, the objectives should be to make cheaper steam (or hot water), and to use less of it.

FUEL ECONOMY—Much of the stress placed on us by the demands of our "all out" effort for national defense production can be relieved by the elimination of waste and by improved efficiency. Outstanding among the materials required for defense work is fuel, and it is therefore of prime importance that those responsible for its use exert every possible effort toward its conservation. Mr. Steiner—of the Ace Engineering Co.—reviews the methods of conserving fuel used for steam generation in oil fired boiler plants; much of the information he gives could be applied to other fuels as well. . . . To make steam at the greatest efficiency, five considerations must be given attention: (1) maintenance of mechanical equipment; (2) firebox maintenance; (3) prevention of firebox losses; (4) control of the combustion process; (5) high rates of heat transfer

Five Considerations

To make steam at the greatest possible efficiency, five topics must be given consideration: (1) maintenance of mechanical equipment, *i. e.*, burners and accessories; (2) maintenance of firebox, (3) prevention of firebox losses; (4) control of the combustion process; and (5) attainment of high rates of heat transfer in the boiler.

It is the function of the oil burner system to prepare the oil for combustion by first bringing it to the proper temperature and pressure and then by introducing it into the firebox in a suitable physical state and mixed with adequate air so that good combustion can result. Each separate piece of equipment in the burner system has bearing on the ultimate success of the combustion process.

Oil Pumps

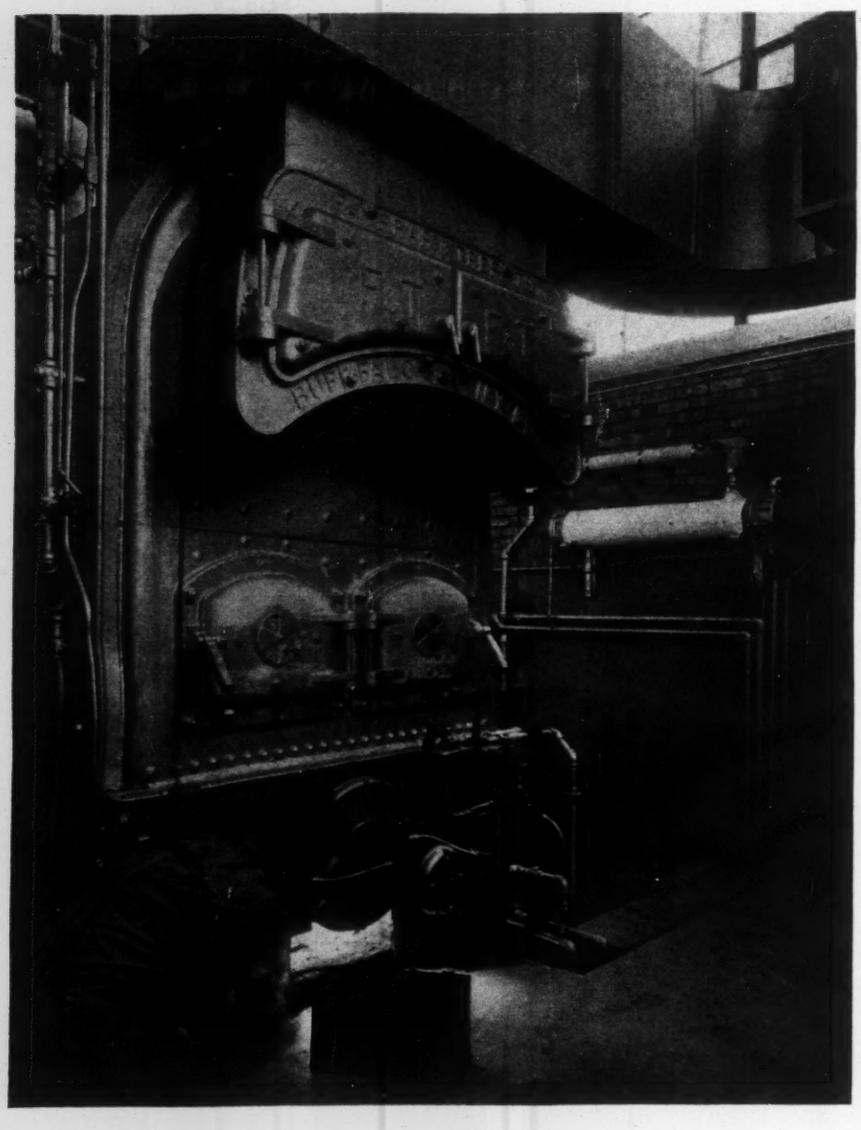
Oil pumps establish and maintain the flow of oil to the burners. Pumps should be kept in good working order. Worn pumps will not keep a constant flow or a constant pressure; therefore worn parts

in the pump should be replaced when necessary. Pump packing, or pump seals, whichever may be used, have to be tight to avoid air being drawn into the oil stream. Air leaking into the pump results in pulsing flow and uneven delivery at the burners. The strainer ahead of the pump is intended primarily to protect the pump from foreign matter borne in the oil. These strainers are important and should be cleaned regularly and maintained in good condition.

Oil Heaters

From the pumps oil passes to the oil heaters. Oil heaters are heat exchangers; the heat transfer rate for which they were designed cannot be attained if the surfaces are fouled. On the oil side, accumulations of carbon can be expected over a period of time, and a well designed heater has provision for cleaning of the tubes. On the heating medium side, a deposit is not likely to form when steam is the heating medium, but when hot water furnishes the heat care must be exercised to prevent heavy accumulations of scale. The subject of scale formation is discussed below at greater length.

There are two important factors dependent upon oil temperature: in many burners the rate of flow through the burner changes with the oil viscosity, which is of course a function of the temperature, and the ability of the burner to break up or atomize the oil effectively is impaired at too low a temperature, also because of increased viscosity. So it is quite important that the heaters do their job of bringing the oil to proper temperature.



Checking combustion and draft should become a daily routine; testing sets need not be expensive, may repay big dividends

Instruments and Controls

The effectiveness at which pumps and heaters are working is usually indicated by gages and thermometers, and the manner in which they work is determined by pressure and temperature regulating devices. Needless to say, the only check on pump and heater performance, other than improper combustion, will be the readings of the instruments, and the only control lies in the accuracy of the pressure and thermal mechanism, so it is only the part of common prudence to verify the performance of these instruments and controllers at regular intervals.

It is worthy of note that many old style burner installations can have marked improvement effected in accuracy of control by substituting modern viscosity compensators for outmoded pressure relief valves and needle valves. One of the points stressed by the Oil Burner Institute, in its pledge of coöperation to Oil Coördinator Ickes, is that the efficiency of older installations will be brought up to present day standards. A good place to begin modernization is in the control of oil feed. Since constant air-oil ratio is fundamental to combustion efficiency, the oil must be metered accurately to start with, or constancy of the fuel-air ratio is hopeless. The matter of air flow control will be mentioned below.

Nozzle or Atomizer

The final handling of the oil is in the nozzle or atomizer. This must be kept smooth and clean. Proper atomization cannot be expected of a fouled or ragged atomizer. Smoke, carbon formation and wasteful combustion are bound to result from neglect of this vital part of the burner system. Should there be any indication that such a part is worn, it should be replaced promptly. Usually, a new atomizer or new nozzle costs only a few dollars—while a drop off in effi-

The subject of fuel economy is not new, but now becomes increasingly important to everyone concerned with running a factory or heating any type of building

ciency of only 1 per cent by reason of poor atomization can in a short time amount to hundreds of dollars. But proper care of the atomizer will assure years of useful service from it.

Oil burners employing fans on the mainshaft for furnishing primary or atomizing air should have these fans, and the associated passages, inspected frequently because grease, lint and other debris settle out from the air and gradually tend to clog the fan and passageways. This restriction rapidly reduces fan capacity and in turn results in smoky and inefficient fires.

The Combustion Chamber

Once the oil has been injected into the firebox, assuming the atomization and air admixture are proper, it remains for the firebox and draft to do the rest. As most operating men know, the principal function of the combustion chamber is to provide a hot environment in which the oil may burn. But another important feature of the firebox construction is the introduction of secondary air (except in the register type burner, where all air is brought in as a single stream surrounding the nozzle). Furthermore, there is often opportunity to use a design that will have the combined effect of insulating the firebox floor and preheating the secondary air. Except for the register type burner, secondary air is brought into the firebox through one or more ports in or near the floor. These ports must be kept clear of carbonized oil and fused firebrick. Checkerwork sometimes becomes overheated and some of the brick or tile break, the pieces falling into the air passages, obstructing and altering the air flow.

When a firebox must be repaired or rebuilt, thought should be given to the possibility of making changes that will improve the combustion. The engineer may find it advisable to consult with a representative of the burner manufacturer to ascertain whether such changes are feasible. Especially will this be true of older installations, for the two reasons that design methods may have changed since the original firebox was built and additional load may have been placed on the boiler since the original installation was made, requiring additional combustion volume. The writer will be glad to give assistance to any engineer who would like advice on this subject.

How Heat Is Lost

The purpose of liberating heat in the firebox is to provide a means of making steam. It is the hope that as much as possible of that heat will pass through the heat transfer surface of the boiler to be absorbed. There are three ways in which heat may be uselessly dissipated instead of being absorbed. The most apparent is the heat carried away in the combustion gases. The second is by transfer to the boiler setting and to the air surrounding the boiler. The third is by cooling of the firebox and boiler water during burner off periods. The first loss can be minimized by proper combustion control, discussed below. The second can be reduced by means of suitable insulation around the firebox refractory and around the boiler itself. The third is a question of burner control.

Refractories

Marked progress has been made in recent years in the development of refractory insulators. There are now many kinds available, to suit almost any operating condition. Some can be exposed directly to the fire, and are in fact insulating firebrick. Some are intended as backing behind the firebrick. Really significant savings can be made in rebuilding old fireboxes by employing some type of insulator.

Air Flow Through Boiler

Many automatic burner installations have been made in heating and process boiler plants under an arrangement that stops the burner at maximum steam pressure and restarts it at minimum pressure. Each time this cycle is repeated a considerable amount of heat can be wasted through the alternate cooling and reheating of the firebrick and the boiler water. Two recommendations are offered to overcome this. Modulating control combined

with the full automatic system will greatly reduce the number of stops and starts; but when the minimum fire still makes more steam than needed, the burner will stop completely, at which time the flow of air through the boiler should be checked by closing of the secondary air port or uptake damper, or both.

Draft Control

The maintenance of combustion efficiency can be summarized in the few words already used above—keeping a constant oil-air ratio. Excess air taken into the firebox over actual combustion requirements in turn is heated from the entering temperature to the stack temperature, and so represents a definite loss of heat. A simple way to check excess air is by means of a barometric draft controller installed on the breeching or at the base of the stack. Other types of draft control are of course available, of greater or less degree of complexity. In the larger installations, the draft control is tied in with the firing control, so that the draft intensity is proportioned to the firing rate, and both are in proportion to the load. Practically every oil burner installation should have a draft control; the smaller jobs will do very well with one of the simpler forms, while in the larger it will generally be found that the fuel saving will be commensurate with the investment in more elaborate forms.

A handy thing around any boiler room is an orsat apparatus for checking CO_2 in the flue gas. Few gadgets will repay better dividends.

Keep Boiler Clean

Mention was made above of the necessity of keeping the oil heater clean, so as to permit of good heat transfer from the heating medium to the oil. The boiler is likewise a piece of heat transfer apparatus, in which the sole objective is the transfer of heat from the combustion gases on one side of the tubes to water on the other side. Interference with this transfer can occur on both sides of the tube—by soot deposit on the fire side and scale deposit on the water side. It might be rash for the writer to remind the engineer of the need to clean the fire side, so the subject will be dropped. The water side is not so obvious to many, being unseen. Nevertheless, scale is often there, and reduces the heat transfer as relentlessly as a soot deposit will. Proper water treatment will prevent such scale formations. [See Water Treatment for Heating Boilers, by Kalman Steiner and Ralph A. Nelson, HPAC, March, 1940, pp. 177-179, and July, 1940, pp. 428-430]. Incidentally such treatment will not only keep the boiler tubes clean, but the preheater tubes as well, should the heating medium be boiler water, as is generally the case with low pressure boilers.

[*In a subsequent article, Mr. Steiner will present a review of the principles of utilizing steam economically.*]

ROUND HOUSE HEATING COSTS CUT

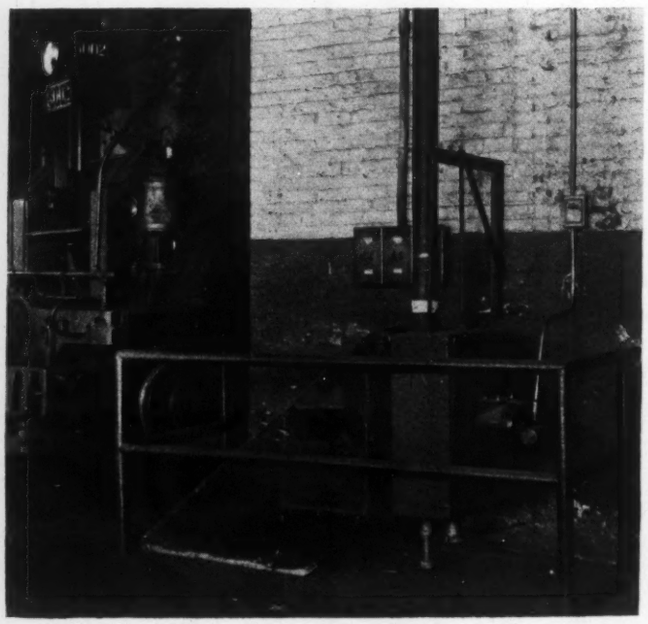
When the Lehigh Valley Railroad's East Buffalo round house was originally built, a central heating system was installed to supply heat and steam pressure to several buildings—the round house itself, a large shop, and storage and hoist sheds. The central heating system had four 300 hp high pressure boilers which, in addition to heating, generated steam to supplement the built-in, forced draft locomotive blowers when fires were started in the locomotive fireboxes. About a year ago the railroad discontinued the use of these separate buildings and moved the shops used for repair, storage and heating into the round house building. Two 125 hp low pressure boilers replaced the high pressure boilers of the old system and 40 unit heaters were installed for heating.

Fig. 1 shows a unique application for a unit heater supplying heat to a bank of five pit stalls. The heat is forced from the unit into a duct running under the floor. There are three units serving this purpose.

In the old system, where steam was used to supplement the forced draft blowers in the locomotives, high pressure was a necessity. With the new installations, an electric blower, suspended from an I beam and attached to the locomotive stack, does the job. Changing from four high pressure boilers to two low pressure boilers, and using unit heaters for heating, has resulted in an estimated saving of $18,000 to $20,000 annually.

[John Kenyon, supervisor of bridges and buildings for the Lehigh Valley Railroad in Buffalo, cooperated in making the above material available. Photo courtesy Fedders Mfg. Co.]

Fig. 1—One of the unit heaters supplying heat to a bank of five pit stalls



Hot Water Heating System Design

New Data on Friction Head in Pipe and Fittings

By F. E. Giesecke, Professor Emeritus, Heating, Ventilating and Air Conditioning, Agricultural and Mechanical College of Texas. Member of HPAC's Board of Consulting and Contributing Editors

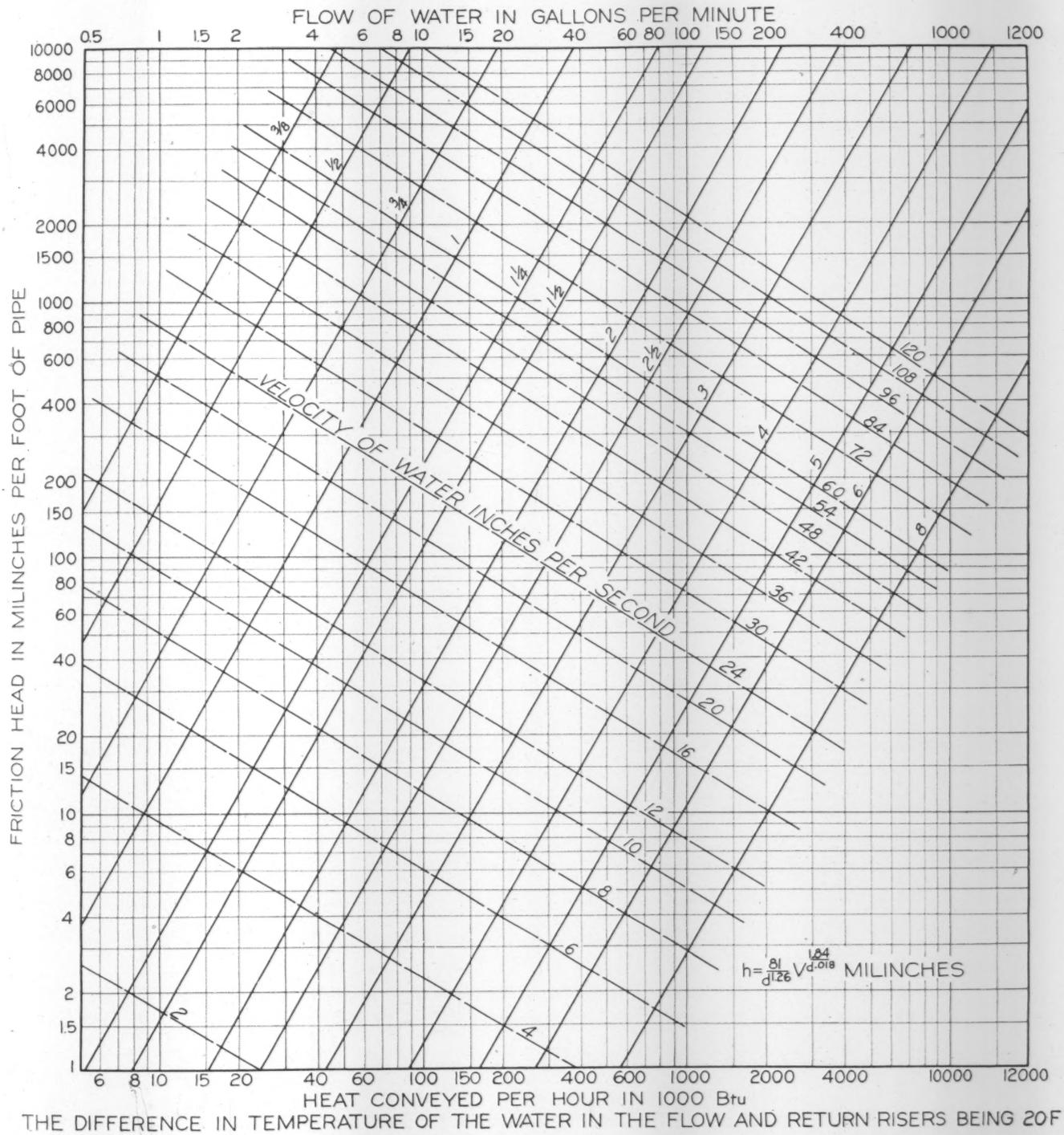


Chart A—Friction head in milinches per average foot of standard weight black steel or iron pipe line. Data represent not friction in a single foot of pipe, but friction in an average foot of pipe line having the normal number of fittings, which affect the friction in the straight pipe for some distance (h = friction head in milinches of water; V = velocity of water, fps; and d = actual internal pipe diameter, in.)

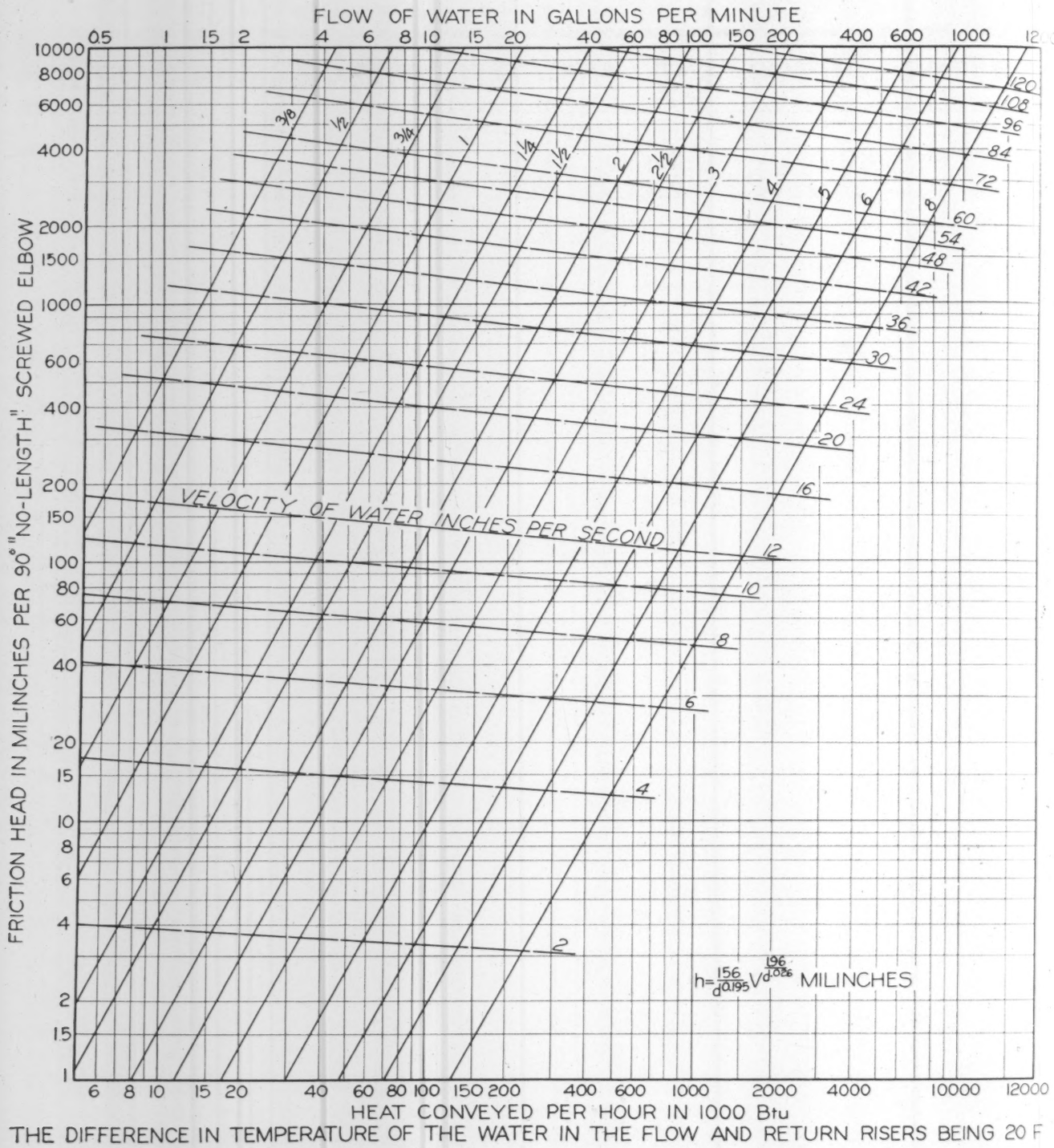


Chart B—Friction head in milinches per 90 deg "no-length" *screwed* elbow. By using "no-length" concept, length of pipe is figured to intersection of center lines of pipe instead of to end of pipe in elbow (h = friction head in milinches of water; V = velocity of water, fps; and d = actual internal pipe diameter, in.) *Note*—Tests for this chart and the one on preceding page were made with 90 F water.

Elbow Equivalents

Relative resistances in number of elbow equivalents

1 90 deg elbow	1.0
1 45 deg elbow	0.7
1 90 deg long turn elbow	0.5
1 open return bend	1.0
1 open gate valve	0.5
1 open globe valve	12.0
1 angle radiator valve	2.0
1 radiator	3.0
1 boiler	3.0
1 tee (100 per cent water deflected)	1.6*
1 tee (75 per cent water deflected)	2.1*
1 tee (50 per cent water deflected)	3.8*
1 tee (25 per cent water deflected)	12.0*

1 tee with branch smaller than run, the number of elbow equivalents for the branch line is approximately

$$0.75\left(\frac{v_1^2+v_2^2}{v_2^2}\right)$$

where v_1 is the velocity of the combined stream and v_2 the velocity of the branch stream.*

*The Loss of Head in Cast Iron Tees, by F. E. Giesecke, W. H. Badgett, and J. R. D. Eddy. Texas Engineering Experiment Station Bulletin No. 41, College Station, Texas. Available on request.

PLANNED SUN PROTECTION . . . .

. . . . . reduces cooling load, simplifies zoning. Les Avery, Avery Engineering Co., cites examples

SUN HEAT—Eight hundred tons is the required air cooling capacity for taking care of the solar load at the Social Security building group in Washington. But the sun load problem is equally important on many small jobs. For example, the room cooler in the bookkeeping office at Toledo Stamping Co. was too small to meet the requirements until sun screen was installed to reduce sun heat gain through the windows. At the Pioneer Linen Supply Co., planned sun protection reduces the load by one-third, enables a 3 ton air conditioning installation to handle the requirements . . . Reducing the sun load through windows is also of value to the engineer in simplifying — sometimes eliminating — the problem of zoning air conditioning systems

Fig. 2—Windows on the second floor of the Pioneer Linen Supply Co. are protected with sun screen

Table 2—Cooling Load Calculations for Pioneer Linen Supply Co.

	Bare Glass No Shading	Venetian Blinds	Sun Screen
Sensible heat gain through walls	13993	13993	13993
Total heat from occupants	4800	4800	4800
Sun load on roof	4862	4862	4862
Outside air	7850	7850	7850
Sun load, glass block	2583	2583	2583
Sun load, glass windows	19600	11400	1960
Maximum Btu per hr	53688	45488	36048

EIGHT hundred tons of air cooling capacity is required to take care of sunshine in the Social Security group of buildings in Washington! That's what W. A. Brown says is the estimated solar load portion of the 4200 ton total.

Mr. Brown—with the Federal Works Agency of the Public Buildings Administration — presented a very able paper at the meeting of the American Society of Heating and Ventilating Engineers held in Washington last summer; his paper was printed in the ASHVE Journal Section in the November, 1940, HPAC. He showed very clearly how the sun load travels around these buildings with the seasons and the hours of the day. There is a total of 146,538 sq ft of glass, some part of which is always exposed to sunshine. Probably one of the largest air cooling installations planned for sun load variation, it is primarily interesting because of its size. But the sun load problem is equally important on many small jobs.

Fig. 1—The room cooler for the bookkeeping office at Toledo Stamping Co. was too small to handle the load until the windows were given sun protection

Table 1—Cooling Load Calculations for Toledo Stamping Co.

	Bare Glass No Shading	Venetian Blinds	Sun Screen
Sensible heat gain through walls	4234	4234	4234
Total heat from occupants	1600	1600	1600
Sun load on roof, 3:00 p.m.	2850	2850	2850
Outside air	980	980	980
Sun load, glass windows	9636	5588	1122
Maximum Btu per hr	19300	15252	10786

In the August, 1940, HPAC I described some tests run in a typical office. Our experience at that time was limited, due to the fact that the material for the article was prepared during early June of 1940. It was our plan to follow up with other more definite tests as time and opportunity offered later in the summer. These are discussed here.

Room Cooler Too Small

Toledo Stamping Co., Fig. 1, has a bookkeeping office facing approximately southeast. There are two windows on the southeast front, and four windows on the southwest side. This is a typical corner office with sun exposure on two sides. Windows had dark colored wood vene-

tian blinds on the inside. The other two sides are interior partitions.

The cooling load for this particular room is shown in Table 1. The sun load is shown for bare windows; with the customary inside venetian blinds; and with sun screen on the outside of the window. The venetian blind data are taken from the 1940 Heating, Ventilating, Air Conditioning Guide, and the sun screen data from tables given in my article in the August, 1940, HPAC.

This table shows that the maximum load, without allowing any reduction in solar load for venetian blinds, is 19,300 Btu per hr, or a little over 1½ tons. With the venetian blinds, there is a total load of 15,252 Btu. Using the sun screen figures, the load is reduced to 10,786 Btu or something less than 1 ton.

This particular room had been equipped with a room cooler which was not large enough to take care of the load during the hot days early in the summer. The owner wished to try sun screen and the installation was completed in August. After the sun screen was installed, the unit had ample capacity to cool the room. Thus the screen saved from 5000 to 8000 Btu, depending upon how you value the protection offered by inside venetian blinds. In this case the sun screen reduced the load to the capacity of the room cooling unit, rated at 10,250 Btu.

Cuts One-Third from Load

Fig. 2 shows the exterior of the Pioneer Linen Supply Co., with the office windows on the second floor protected by sun screen and Fig. 3 shows the interior of the main office, in which a 3 ton air conditioning unit is installed with ducts to private offices on the west side. Table 2 gives the Btu load calculated for this job, from which you will notice that figuring the bare window glass, without any protection, it totaled 53,688 Btu or about 4½ tons cooling load. The reduction with polished aluminum blinds on the inside of all south and west windows makes the load 45,488 Btu, and with sun screen the load is 36,048 Btu—practically 3 tons.

In this particular installation, the temperatures inside these offices before air conditioning and the sun screen was installed would rise 10 to 15 F above the outside temperature when the sun was shining. The office faces south, and the worst load came about 3 p. m. The 3 ton machine was considered ample, and the results have proved it to be. It was a simple matter for this size unit to hold the temperature at 80 F during the hottest weather, and there has been no panel heating effect from the windows to make the employees near them uncomfortably warm.

Solves Zoning Problem

Reducing solar heat gain through windows is particularly important where the sun exposure is such that zoning would be required. At the Pioneer Linen Supply Co., for example, the end office, Fig. 4, has both a south and a west window, and ordinarily zoning would have been required. By using sun screen on both of these large windows, special zoning was unnecessary. The temperature stayed within 2 F of the temperature in the main outer office that had southern exposure

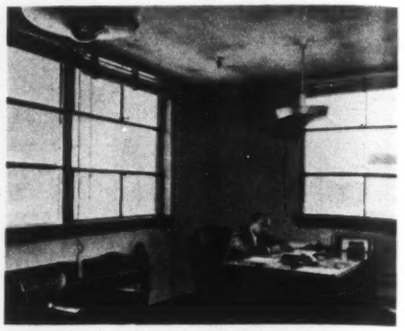


Fig. 4—This office would present a zoning problem were it not for the reduction in solar heat gain from sun protection

only. To the air conditioning engineer, this means a simple solution of a difficult problem—the problem of moving the cooling load around the building to follow the sun load.

Sun Intensity Varies

Any method of sun protection must be rated as of the hour at which it is being considered. In other words, the sun intensity varies from sunrise to sunset. It also varies depending upon the angle of incidence to the window exposure in question, both in the horizontal and vertical plane. It is incorrect to say that any device, whether it be sun screen, awnings, venetian blinds, heat absorbing glass, glass block or what not, passes any fixed percentage of sun load. The heat absorbed or reflected by these materials is variable, depending upon the angle of the sun with relationship to the window.

The designing engineer will, therefore, check this sun exposure carefully to determine the time and quantity of maximum solar load. This will indicate the savings possible if planning sun protection.

Fig. 3—A 3 ton air conditioning unit handles the load at the Pioneer Linen Supply Co., but would be too small without planned sun protection

New catalogs, bulletins, etc. are reviewed in the department "Trade Literature" published in HPAC monthly

ON THE UP AND UP

. . . . is the demand for air conditioning at the Merchandise Mart, says W. A. Stahl, Operating Manager

WHEN the Merchandise Mart, Chicago, was erected 11 years ago only two tenant spaces were air conditioned. The National Broadcasting Co. studios occupied 76,211 sq ft on the 19th and 20th floors and their conditioning was accomplished through the medium of a 170 ton centrifugal machine in the boiler room of the building. Chilled water was supplied to the fan room on the 20th floor through a 6 in. chilled water riser and return. This installation was supplemented in 1935 by the addition of two 25 ton compressors on the 19th floor, which served an additional 5000 sq ft of studios and offices. This second fan room was equipped with direct expansion coils. Both compressor installations were cooled by the use of city water. Every attempt was made in the NBC installation to do as complete and perfect a job as was afforded by the art at that time.

The main building restaurant and coffee shop, occupying 34,756 sq ft of area, was cooled through the medium of a CO_2 machine at the track level of the building supplying direct expansion coils in the fan housing just above the restaurant. The restaurant duct system was served by one large fan supplying 70,000 cfm of air, with no attempt made at zoning and with a minimum of controls. The operation of the restaurant system, until changes were made this year, was never wholly satisfactory, it being impossible to hold proper temperatures or humidities.

The Chicago River supplies condensing water for the refrigeration machines. The water is returned to the river 60 ft downstream from the intake

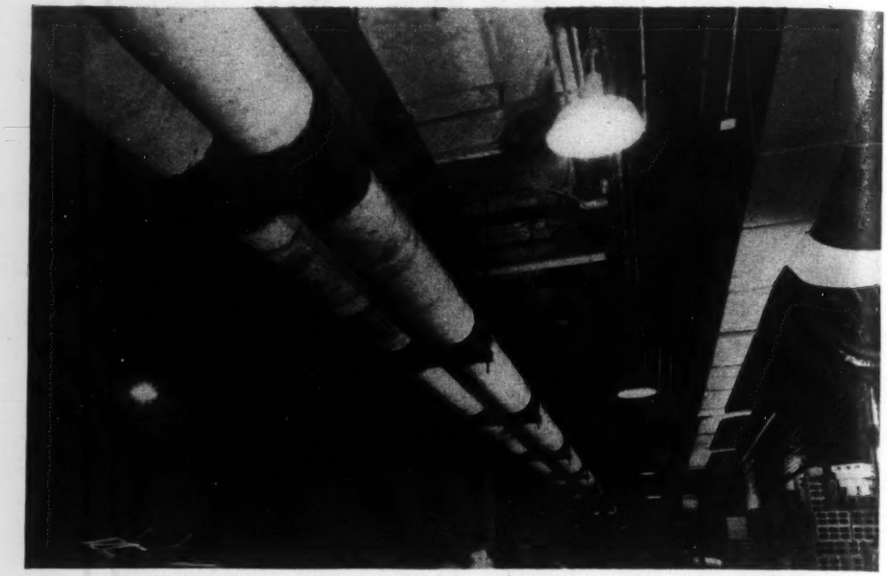
When the building was constructed, no ventilation was supplied the first seven floors, except for the stores and restaurant on the first floor. All of these floors were intended for warehouse space and were furnished with exhaust only. All of the upper floors were provided with ventilation from two fan rooms, one at each end of the building, from which outside air was blown through ducts for the entire floor (210,000 sq ft).

Growing Need for Air Conditioning

In 1936 the building management felt that there would be a growing need for air conditioning in the show rooms, offices and stores in the Mart. It was, of course, considered desirable to plan for a central system to supply cooling where needed instead of allowing the installation of less efficient isolated units throughout the building. A large number of self contained units were thought difficult to service and to present a noise and flooding hazard. Another disadvantage of individual units in the Mart is that the building was laid out with very few wet stacks. Each floor has four large toilet rooms for the use of tenants, so that stacks to provide for individual lavatories in tenant spaces are unnecessary.

This welder, if working alone, would have to spend all of 1941 on welding the chilled water lines for conditioning the Merchandise Mart

UP AND UP goes the demand by tenants for air conditioning service at the Merchandise Mart, mammoth Chicago building. When the building was constructed 11 years ago, only two tenant spaces were air conditioned. Many installations have been put in since then, and many more requests than were anticipated have been made this year . . . . . Jake Stahl describes the development of air conditioning at the Mart, and the plans that have been made to provide a service which almost daily becomes of ever-increasing importance to tenants and management



Six hundred feet of 16 in. chilled water piping with 2000 in. of welded joints

Condensing Water from River

A start was made on this central plant in 1937 with the installation of one 350 ton centrifugal machine. While the original centrifugal machine, installed for NBC cooling when the building was erected, used city water for cooling, the new installation was designed to take advantage of the proximity of the Chicago River, along the south front of the building, for its cooling water. It was decided to set aside sufficient space at the track level of the building for cooling machines, together with auxiliary pumping equipment, etc., for the entire structure. The original installation provided for a 14 in. water line and return to the river, this being sufficient for approximately one-third of the total building. The river water is taken in through a crib which is of sufficient size to accommodate all condensing water needed for cooling the entire structure. The tunnel through which this cooling line is run is also large enough to accommodate such additional capacity. Water is discharged back into the river at a point 60 ft downstream from the intake.

Warehouse Areas Become Finished Space

About the time that the central refrigeration plant was started, demands for space were such that it became necessary to develop warehouse areas on the lower seven floors into finished space for the accommodation of new tenants. It was decided that these new finished spaces would be equipped for ventilation, sizing all duct work properly to accommodate future air conditioning for these spaces. It would then be necessary only to furnish chilled water coils in the fan rooms, together with a supply to these coils to cool any of these new spaces. The first space cooled by the new refrigeration plant was a section of about 27,000 sq ft on the seventh floor (former warehouse floor). This installation was completed during the summer of 1937.

In 1938 conditioning was installed in the Merchants and Manufacturers' Club (where, incidentally, the Illinois chapter of the ASHVE meets). The club was built in previously unfinished space and with the duct work especially designed for air conditioning. During this same year we reaped the first benefits of planned ventilation for future air conditioning. A tenant occupying 90,000 sq ft of space served by four fan rooms asked that air conditioning be installed. This installation was accomplished without the necessity of any changes whatsoever within the tenant area, it being necessary only to add the chilled water coils and controls in the fan room which we served from our central plant.

Requests Increase

The Mart now began to receive requests from certain tenants for cooling on upper floors where spaces were ventilated by duct work which was not applicable to air conditioning. Very often other tenants in the area to be serviced did not yet desire air conditioning. Thus, while cooling from central fan rooms was desirable, it was necessary to plan for these tenants who desire conditioning previous to others in their immediate vicinity. It was decided, therefore, to serve these tenants with chilled water units supplied from our central plant, the units in turn furnishing air for individual duct systems.

In 1939 air cooling systems already installed and requested were such that it was necessary to install another centrifugal machine. A machine of 525 tons was installed, together with additional river and chilled water pumps.

In 1940 again, a number of new areas were conditioned, among them being installations in one of the world's largest drug stores and also the Merchandise National Bank. These two installations, together with the first floor stores, completed the air conditioning of the finished spaces on the first floor of the Mart. At the end of 1940 we had sold practically all of the capacity of the two building machines and it was again necessary to budget for another centrifugal machine for 1941.

In the first part of this year the main building restaurant decided that its cooling system was out of date and inadequate as compared to the systems that had been recently completed in other spaces on the first floor and an arrangement was made to trade in the equipment and serve the space from the central plant. It was necessary to revamp all of the coils in the fan room to enable them to use chilled water as a cooling medium instead of direct expansion. Recirculating ducts were also installed; previously, this system operated on 100 per cent outside air.

New Centrifugal on 20th Floor

Early this year the NBC decided to enlarge their studios and offices. As the cooling equipment was operating at full capacity, this meant

the addition of another compressor for cooling. The tenant requested that, instead of the new compressor, we consider trading in the two compressors then in use on a new centrifugal to serve the entire area, and the proposition was finally worked out on that basis. However, instead of installing the new machine in the boiler room 400 odd feet below the studios, it was decided to place the machine on the 20th floor, thus cutting out the losses due to the long chilled water lines and making for more efficient operation from a personnel standpoint.

An interesting thing about the new installation is that, while the new centrifugal machine is of 275 tons capacity as compared to the 170 tons of the old installation, the 250 hp motor and gear of the old machine were used to drive the new one. Also, because these machines are now constructed of steel plate instead of castings the weight of the machine is not in excess of the floor loading available.

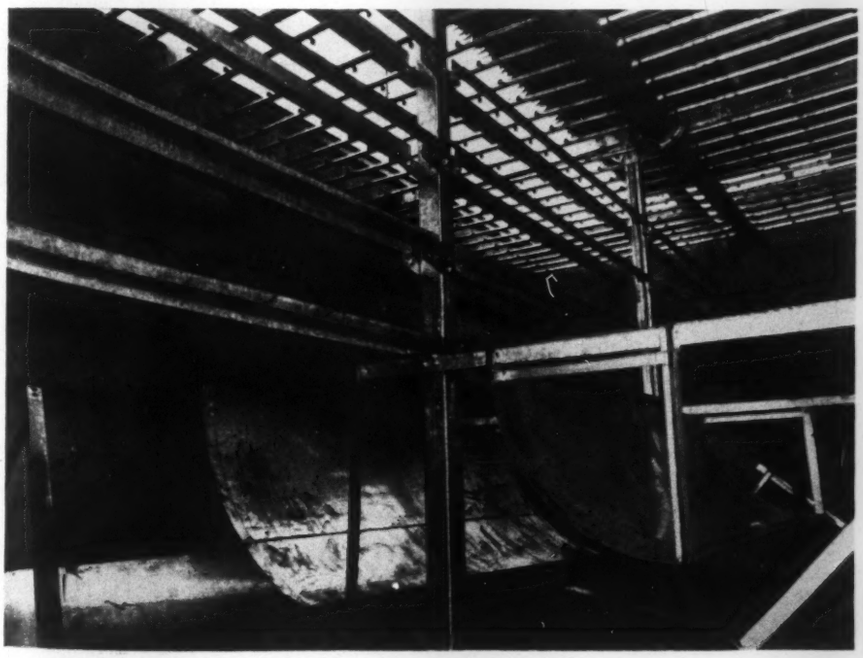
It was not found practical to use river water for cooling the new machine as 8 in. pipe lines in excess of 800 ft and return would have been required. Also, inasmuch as this unit is entirely under the tenant's supervision, it was not practical to tie in the operation with that of the machines operated by the building. It was decided, therefore, to install a cooling tower on the roof directly above the machine. Since this tower is visible from all of the upper offices of the Mart, it was constructed of brick to match that of the penthouse on the 21st floor. Two fans supply 100,000 cfm through the tower.

Supplementary dry type filters are used behind all viscous filters in the Mart cooling installations. There are also several installations of combination electrical precipitation and automatic viscous self-cleaning filter units in the building

Another part of the conditioning program for 1941 is further completion of the system of chilled water risers. The riser originally installed was 14 in. The installation of a 16 in. lateral to the west end of the building which is to serve 10 in. and 12 in. risers at typical fan room locations has just been completed. Due to the difficuly of obtaining cork, the new 16 in. lateral pipe was insulated with cork at the hangers only, with double ¾ in. fill in between.

Future extension of Mart air conditioning will, of course, be guided by tenant demand. This year there have already been many more requests for cooling than we anticipated.

A brick cooling tower — shown here under construction — was installed to supply condensing water for the refrigeration machine on the 20th floor



Water for Industrial Uses—One phase of the work of the American Society for Testing Materials which is relatively new—at least in comparison with the work under way in other fields—is that on water for industrial uses. The Society's committee D-19 has during the last two years stimulated the presentation of a great deal of data and the discussion of test procedures. Standards already developed include methods of sampling plant or confined waters for industrial uses, method of reporting results of analysis of industrial waters, and a method for determination of the hydroxide ion in industrial waters.

Interest at the ASTM's recent meeting was concentrated on the symposium on problems and practices in determining steam purity by conductivity methods, which comprised six technical papers.

Following an introduction by R. E. Hall, who acted for the committee with the chairman, Max Hecht, in developing this important contribution, Messrs. Belyea and Moody discussed the sampling of steam and boiler water.

Dr. D. S. McKinney, who has participated actively in many phases of committee work, covered the calculation of corrections to conductivity measurements for dissolved gases.

B. B. Place, in his paper on the Degasification of Steam Samples for Conductivity, which constitutes a summary of the methods and apparatus available for the elimination from steam samples of gaseous impurities that affect conductivity measurements, pointed out that only acid and basic gases which combine chemically in solution to give ionized compounds affect conductivity measurements.

A. R. Mumford presented a paper covering a New Type of Conductivity Apparatus for Use with Boiler Waters and Steam Samples, and C. E. Kaufman discussed Conductivity Cells and Electrical Measuring Instruments.

Production for National Defense

AIR CONDITIONING A NECESSITY

E. E. Herbacek Describes Year 'Round Zoned System for New Defense Plant

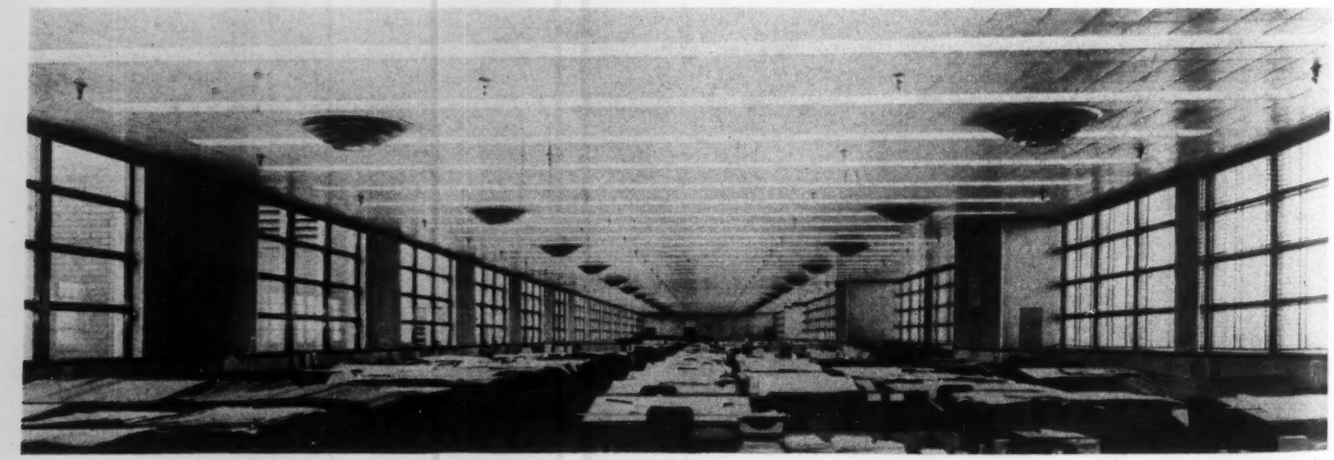
AT FRIDLEY, MINN., stands the newly completed plant occupied by the Northern Pump Co., consisting of several manufacturing buildings 1200 ft long fronted by an attractive administration building. This project was completed and in manufacturing operation in record time of less than four months from the date when ground excavating operations were started. The administration building is two stories high, 421 ft long and 40 ft wide; it houses the Northern Pump Co. and U. S. Navy executive personnel and the engineering, designing and general office forces.

A NECESSITY — Air conditioning is playing a vital rôle in production for national defense. Aside from its myriad uses in providing proper temperature and humidity conditions for processing various materials, an important function it fulfills is the maintenance of the efficiency of our greatest resource—manpower—through the creation of comfortable working conditions for executives, engineers, draftsmen, office workers and others who must perform their tasks under the stress of our "all-out" effort . . . . . E. E. Herbacek, chief engineer and secretary of the Spencer Air Conditioning Co., describes the automatically controlled, zoned, year 'round air conditioning system for a defense industry plant

The building is brick with large stationary windows of double hermetically sealed glass. The outside walls and roof surfaces are insulated and sealed to reduce heat leakage and prevent condensation of moisture on the exposed surfaces during the winter months. The location of the building and its exposure to extreme outdoor weather conditions necessitated installation of a complete and automatically controlled split system for heating and year 'round air conditioning.

Heating of the building during winter months is accomplished mainly by a system of convector type steam radiation installed throughout

The engineering and drafting room. Air conditioning has been found to increase markedly the efficiency of such departments



Looking between two air conditioning units toward the refrigerating machine. The fans shown handle return and exhaust air

the entire building and supplemented by booster heating, ventilation and humidification through the main air conditioning system.

During summer operation the air conditioning system provides cooling, dehumidification and ventilation. Air distribution throughout the building is accomplished by ceiling type diffusers of the supply and extract design which are mounted in suspended ceilings that conceal all the ducts and also provide spaces above the ceilings over the entire building which are used as plenums for return and exhaust air through the extract portion of the air diffusers.

Zoned Air Conditioning

The air conditioning equipment room is in the basement of the center portion of the building, and due to the extreme length of the building, together with the limitations in ceiling heights, it was necessary (and also desirable from a standpoint of operation and control) to divide the air conditioning into two separate systems, each of which takes care of half of the building from the ends to the center.

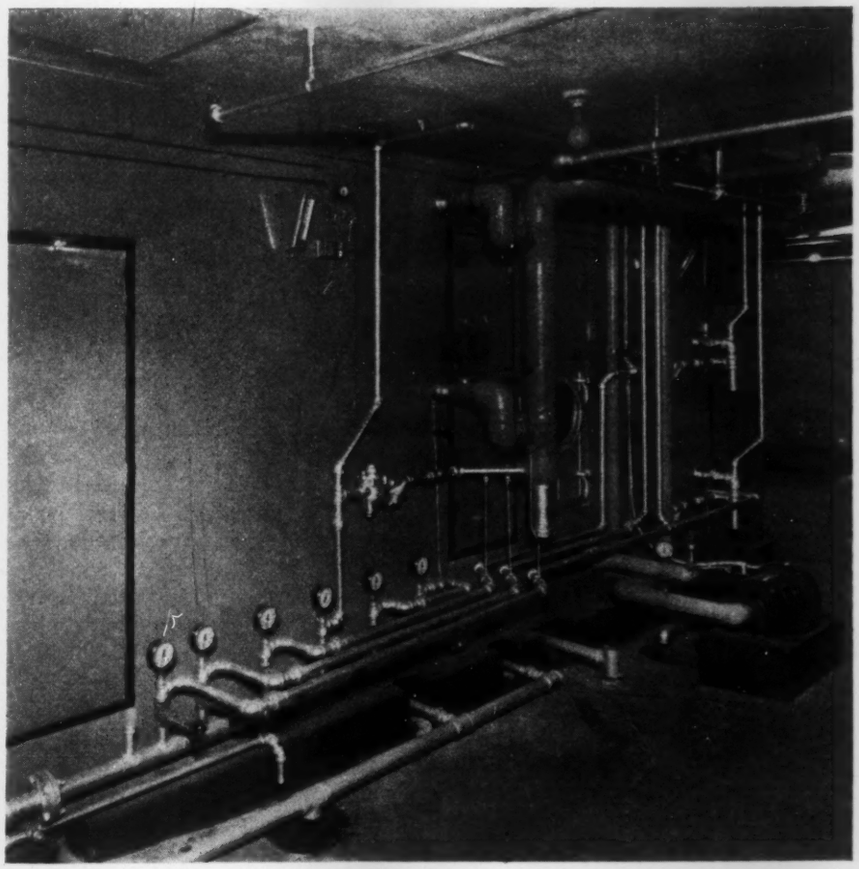
It was necessary also, due to varying solar heat exposures and internal heat load conditions, to split the two air distributing systems into seven separately treated and controlled zones so as to maintain uniform and comfortable conditions throughout the building, regardless of varying outdoor conditions.

Each of the two air conditioning units consists of cleanable type air precleaners, preheat coils, air washer and humidifier, reheat coils, main air supply fan, main return air and exhaust fan, and direct expansion zone cooling coils connected to a 100 hp "Freon," eight cylinder, fully automatic refrigerating machine.

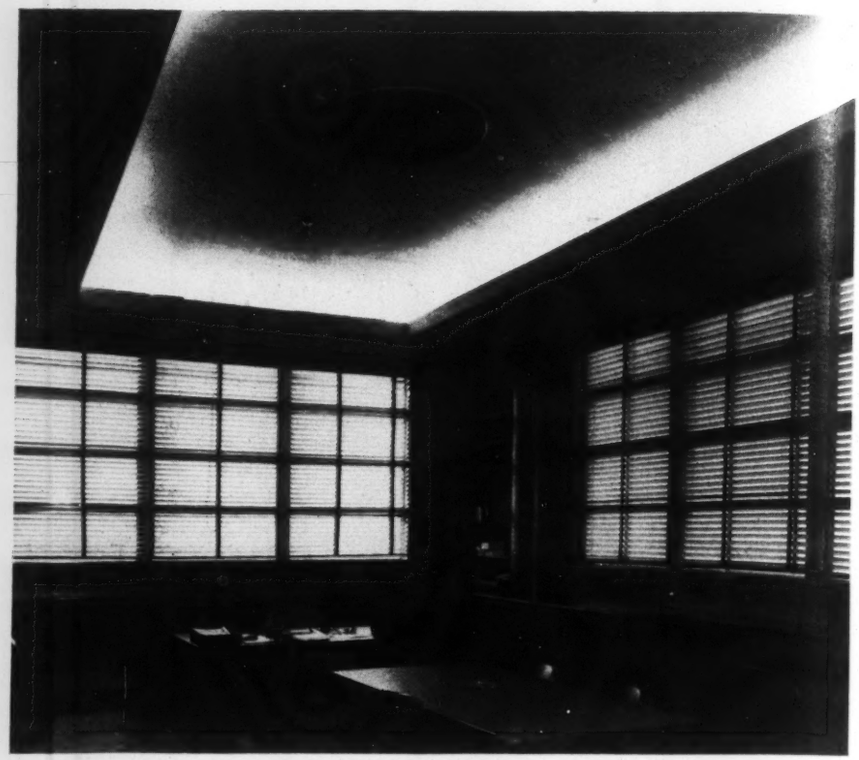
Moisture eliminators and drain pans are installed at the leaving air side of each cooling coil for trapping the entrained moisture resulting from dehumidification of the air. A system of waste condensate piping is connected to all of the eliminator drain pans and sump pits.

Water Pond on Roof

A portion of the condensing water required by the refrigerating



Front view of one of the air conditioning units, showing steam piping to the tempering and reheating coils, water pump and piping to the air washer. The end of one of the cooling coils and piping can be seen at the upper right hand corner



A private office, showing the supply air diffuser mounted in a coved ceiling. Return and exhaust air is vented through slotted openings hidden by the edge trim

machine is piped to the roof of the building for maintaining an evaporating surface to reduce the solar heat leakage into the building. The level of water is maintained by adjustable standpipe fittings installed in the rain water leaders at the roof.

Two speed motors are provided for each of the four fans to vary the volume of air supply required during winter and summer operation, as the cooling load requirements demand a greater volume of air during summer operation than is required during winter operation.

Each of the two units and systems has a maximum capacity of 32,500 cfm of air, making a total of 65,000 cfm and requiring 108 tons of refrigeration for cooling.

Air Circulation

Manual adjustable louver dampers are installed in each of the seven zone supply air mains for proportioning the volume of air required. The air conditioning units and all ducts within the equipment room are insulated with cork and finished with asbestos cement and canvas jacket. In addition, the main supply and return air ducts are lined with acoustic sound insulation. Insulated brick air shafts are provided on the outside rear wall of the building and connected into the units for outside air intake, return air and exhaust air, and each of these stacks is sized for 100 per cent of the unit air capacity.

All air supplied to the building (except that supplied to the blueprinting, photostating, photograph and rest rooms) is drawn back through the return air system and fan, and can be all recirculated through the system or all exhausted or partially recirculated and exhausted, depending upon the amount of outside air being introduced. A portion of the exhaust air is supplied to the basement equipment room for removal of heat generated by the fan and refrigerating machine motors. This air is vented at the extreme ends of the basement through vent stacks connected into gravity roof ventilators.

Pneumatic Control

A complete system of pneumatic automatic temperature and humidity control is installed to maintain proper year 'round air conditions in the seven separate zones of the building served by the air conditioning system. In addition to the air conditioning control system, each steam radiator is automatically controlled. Automatic proportioning type outside air, return air and exhaust air dampers regulate and proportion the amount of and mixture of air distributed by the system. Humidification is accomplished by a single stage air washer operated with a dew point control actuating motorized proportioning steam valves on the preheat coils.

Dehumidification is accomplished by direct expansion refrigerant in the seven zone cooling coils situated in the supply air plenum chambers. Low limit supply air controllers are installed in each system to deliver air to the building at ventilation temperatures during the off-heating cycle. Seven zone cooling thermostats actuate electric solenoid refrigerant liquid stop valves on the zone cooling coils for control of cooling temperatures. Automatic relays actuated through the fan motors close the outside air dampers and refrigerant valves on a shut-down of the system.

The outdoor compensating control system consists of master compensating instruments which determine the outdoor temperature and automatically actuate sub-master zone thermostats to maintain a maximum inside temperature differential. As the outdoor temperature drops, the inside differential decreases to a point where cooling is cut off and heating is cut in.

The refrigerating machine is equipped with capacity unloaders actuated automatically by the cooling demand through the zone cooling coils.

Exhaust Systems

In addition to the main air conditioning system, there are installed separate systems of exhaust ventilation to take care of the toilet and rest rooms. These systems are connected to exhaust fans in the penthouses on the roof of the building. Exhaust systems are also provided for the blueprinting, photostating and photography departments, as air supplied to these spaces is not returned to the main system.

Numerous air systems are installed throughout the factory buildings for exhaust ventilation from toilet and locker rooms and other factory spaces, and there are also systems for vapor exhaust and dust control and elimination.

New Method of Duct Construction SIMPLIFIES AIR CONDITIONING

J. M. Cunningham Describes Fabrication of Ducts Which Do Not Need To Be Concealed

DUCT CONSTRUCTION—To overcome the difficulty and expense of concealing air conditioning ducts, a method of construction has been used on several recent installations such that no seams show on the outside. The duct presents a smooth surface throughout, and when painted to match the ceiling appears to be an integral part of it. . . . . Mr. Cunningham, Carrier Corp., presents construction details and recommendations and explains the advantages of the method

THE difficulty of concealing air conditioning ducts is a problem with which every architect and consulting engineer has had to contend. To overcome this obstacle and eliminate the need for it, a type of duct construction* has been used for several recent installations such that no seams show on the outside. The duct presents a smooth surface throughout and when painted to match the ceiling, appears to be an integral part of it. The unsightly effect characteristic of exposed ducts is thereby eliminated.

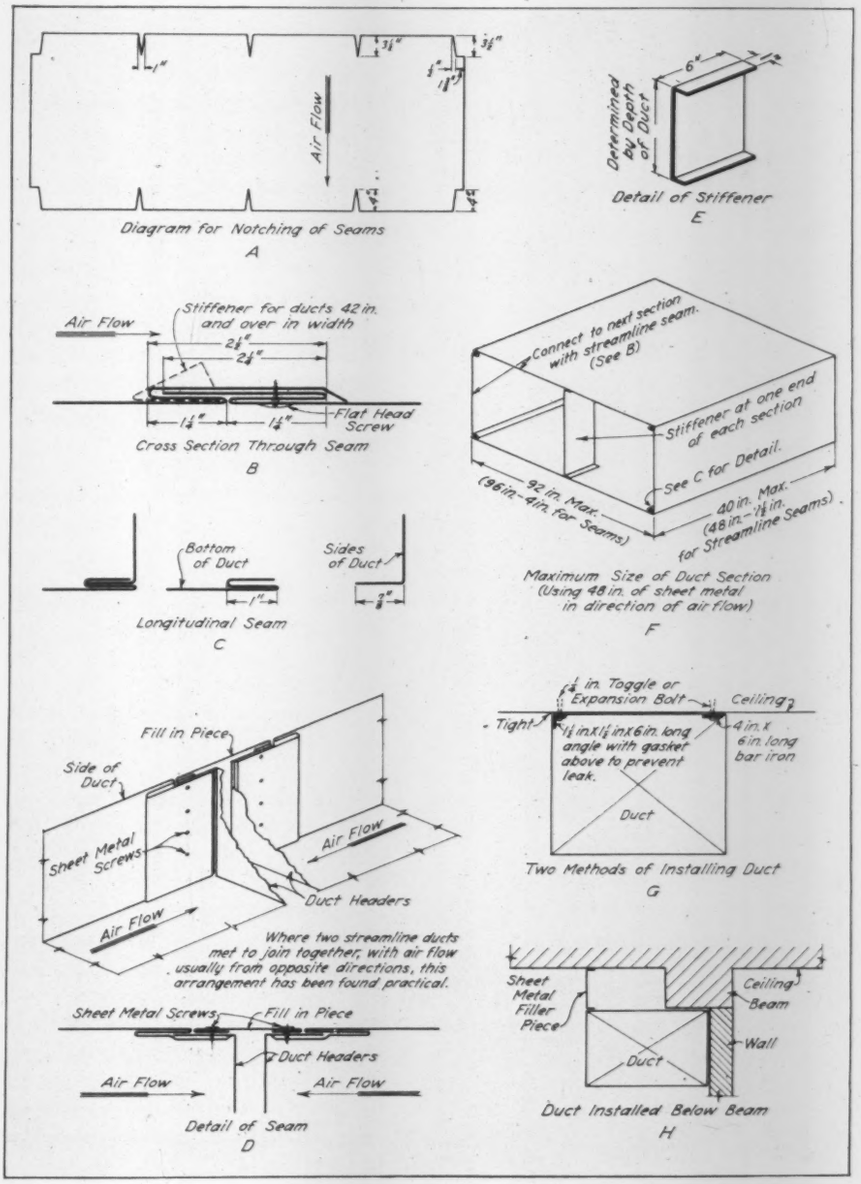
Method of Construction

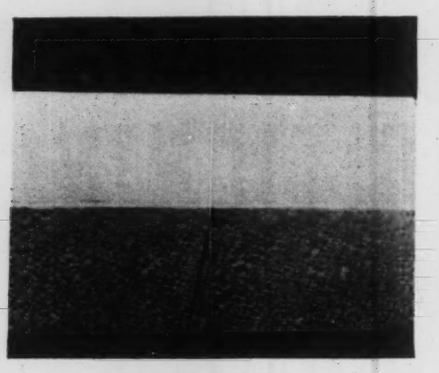
Construction of these ducts introduces many changes in sheet metal practices, as illustrated in the sketches. Notched along the seam edge (*A*), the sheets are folded into a seam (*B*) for joints in straight runs of ducts. The seams are held together with flat head sheet metal screws. The holes through the seams are drilled. The screws are counter sunk so that the screw heads do not protrude above the sheet metal. The lapping of sheet metal for longitudinal seams at the corners of the ducts (*C*), held tight by screws as in the simple seam, assures a smooth and firm finish at all corners. The seams and joints of duct fittings, as for offsets and elbows, are inside the duct, spot soldered and smoothed off with a rasp.

To avoid the appearance of gaps between the ends of two duct head-

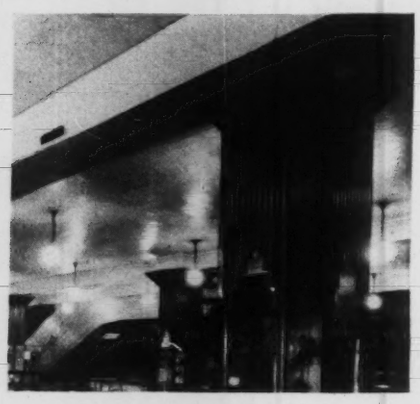
*Patent applied for.

Construction details for ductwork presenting a smooth surface throughout, which may be painted to match the ceiling and does not require concealing


Air Flow
Diagram for Notching of Seams
A
Determined by Depth of Duct
Detail of Stiffener
E
Air Flow
Stiffener for ducts 42 in. and over in width
Flat Head Screw
Cross Section Through Seam
B
Connect to next section with streamline seam. (See B)
Stiffener at one end of each section
See C for Detail.
92 in. Max. (96 in.-4 in. for Seams)
40 in. Max. (48 in.-1½ in. for Streamline Seams)
Maximum Size of Duct Section (Using 48 in. of sheet metal in direction of air flow)
F
Bottom of Duct
Sides of Duct
Longitudinal Seam
C
Fill in Piece
Side of Duct
Sheet Metal Screws
Air Flow
Duct Headers
Where two streamline ducts met to join together, with air flow usually from opposite directions, this arrangement has been found practical.
Sheet Metal Screws
Fill in Piece
Duct Headers
Detail of Seam
D
¾ in. Toggle or Expansion Bolt
Ceiling
Tight
1½ in. x 1½ in. x 6 in. long angle with gasket above to prevent leak.
4 in. x 6 in. long bar iron
Duct
Two Methods of Installing Duct
G
Sheet Metal Filler Piece
Ceiling Beam
Wall
Duct
Duct Installed Below Beam
H



Sample duct of the new construction, showing one seam unfinished and the other filled with compound and painted



Close to the ceiling so the top is not a dust catcher, and painted to blend with it, the duct is inconspicuous

The duct is painted, looks like a beam

Ducts may be painted and grained

ers, a fill in piece (*D*) is used to bridge the space between the two ducts. The continuous external surface lends itself more readily to room interior decorations than one in which gaps occur.

In some sizes of ducts a stiffener (*E*) is required to keep the construction rigid; it is located inside the duct (*F*) so that the face of the stiffener is parallel to the air stream and causes no appreciable increase in resistance to air flow.

There are several methods of supporting the ducts (*G, H*). When hangers are necessary in supporting the duct, they should be located inside it.

Recommended Gages

Recommended gages of sheet metal for the various duct sizes are:

Duct with dimension 15 in. or less	26 ga
16 in. to 41 in.	24 ga
42 in. to 60 in.	22 ga
Maximum dimension 61 in. or more	20 ga

A careful selection of the sheets is required to assure a smooth surface of the exposed ducts. Flat sheets 48 in. x 96 in. are recommended for use to obtain the best results in final appearances.

The cement or putty compound to smooth across the seams must adhere to the metal, and must take a good paint finish. It is recommended that the compound have a rubberized base and be insoluble in water. The compound should be applied in a thick coat and smoothed over the seams and screw heads with a putty knife or small trowel.

Advantages

The advantages of this new method of construction have been found to outweigh the additional cost; with more experience and increased knowledge of design and installation details, sheet metal mechanics will be able to reduce the cost by shortening the time required for construction and erection.

Furring in is not necessary for concealment. By eliminating standing seams, the ducts can be located next to the ceiling or beam, doing away with the clearance at the top of the duct which is usually such a dust catcher. In rooms with low ceilings, the ability to locate the duct at the ceiling (instead of an inch or so below) often means an appreciable and worthwhile saving in head room.

INDUSTRIAL RESEARCH A NATURAL RESOURCE

Naming industrial research one of America's greatest natural resources, E. R. Williams, manager of the Johns-Manville research laboratories, told the 75th anniversary science meeting of the University of New Hampshire recently that if such research is encouraged and supported properly, the United States will not have to accept government regimentation as Germany did.

"With the United States setting a record in technical manpower, research laboratories and engineering facilities, it lies within our power to create and provide the personnel for the greatest engines of liberty and freedom ever created."

As examples of "modern miracles" turned out by research laboratories, Mr. Williams cited the work done by the petroleum industry in producing synthetic products such as 100 octane gasoline for airplane fuel; synthetic rubber; toluene, essential to TNT and other explosives required in wartime; glycerine; and other basic industrial chemicals.

"As a result of industrial research in the petroleum field, the yield of gasoline from crude oil and the production of lubricating oil both have doubled," Mr. Williams said. "Stocks of gasoline are eight times as large as in 1917, while those of lubricating oils are more than two times as large, and of fuel oils, 10 times as large. The unused refinery capacity in this country of around 1,000,000 barrels daily is greater and more efficient than was our entire capacity in 1918."

Past accomplishments of industrial research must be considered in the light of what part they can play in defending America and the great defense effort, he said, and cited the insulation of heated equipment in industry, which is now accounting for a great amount of fuel conservation.

"The company that prepares for the future, the one that can give the customer more for his dollar, and add to the comfort of humanity, is the organization most likely to survive and achieve the success for which we all are striving," Mr. Williams declared. "Upon this type of industrial thinking the future of America depends."

"Open for Discussion"

Communications to the Editor

TESTING AND DRYING REFRIGERATION SYSTEMS

THE EDITOR—

I have read thoroughly Mr. Ramsey's article on Testing and Drying Refrigeration Systems, which appeared in the June HPAC. It is quite evident that the author has been confronted with the many problems incident to the proper dehydration of and the bad after effects of improperly prepared "Freon" systems. The article shows also a good practical knowledge of the subject. Therefore, any remarks which I might make should only be construed as being helpful additions, time-saving suggestions and possible improvements.

It has been my experience that soap and water for testing is effective only for large leaks and rather time-consuming. It is slow work to cover properly a leak with a soap film so that a bubble is formed. Since but a small trace of the halogen refrigerants will give a test with a torch, why not add a small quantity of "Freon" to the air used for obtaining pressure during the testing and find all of the initial leaks large and small, even to porosity of castings (not uncommon), at one time with a torch? No system properly erected has leaks so large that there is danger of losing all of the test air and "Freon." Again, there is no unsightly film left when the soap dries and you don't have to test and retest.

Has the author ever considered cooling the exhaust air from his air compressor, by passing this into a receiver constructed like a water cooled condenser? This is done before allowing it to enter the system to be tested. A water cooled receiver will reduce the compressed air below room temperature and take out additional moisture. The dryer the air entering the system, the easier will be the job of dehydration later, since the dryer air will pick up more moisture. It is a simple matter to bring cooling water to such a receiver with high pressure hose. A further precaution would be to attach the air compressor suction to a large dryer.

Mr. Ramsey mentions closing valves dividing the system. This is good; however, valves have a habit of leaking at the wrong time and the use of air with "Freon" in it when pumping up pressure makes this unnecessary unless a leak is found which necessitates making repairs at atmospheric pressure on the leaky section after repairing. It is only necessary to admit air and "Freon" mixture to that section for testing, by opening a section valve and again raising the pressure in the whole system with compressed air. "Freon" is costly and a saving can be made in this way.

It is assumed that in indicating a loss of 2 psi pressure over a period of seven days that corrections for ambient temperature were made; however, whether this amount is large or small depends on the total volume of the system. A leak, however, is indicated which could be located with air and "Freon." What does Mr. Ramsey do when he is not so fortunate in having so much time for testing? The air-"Freon" mixture will shorten the testing time.

The make of equipment used on the installation is not mentioned and, therefore, it is impossible to tell what care has been taken in manufacture to eliminate free moisture in the pores of the metal, in pockets, etc., which may be under a heavy oil film. Experience has indicated that the vapor pressure even at high vacuum may be insufficient to break through the oil, thereby leaving considerable free moisture in a system. As a consequence, it seems advisable unless equipment has been baked in 190 F while a vacuum of 0.2 in. of mercury absolute is being applied, in manufacture, to augment the field dehydration by vacuum with a liquid line dryer, either of a temporary or permanent type, for safety. Moisture, even in small amounts, in a "Freon" system can cause a lot of trouble later; depending of course, upon the amount present. The less water the longer the trouble is delayed and it is best to use every precaution. A moisture content of less than 0.003 per cent is a good figure to strive for, whether it be an air conditioning or refrigeration system. If this goal is attained no serious trouble from hydrolysis will result.

I have enjoyed reading the article and hope that my remarks may help to further sound practices.—H. R. JAEGGLI, manager, service dept., Carrier Corp.

THE FUEL OIL SITUATION ON THE ATLANTIC SEABOARD

THE EDITOR—

The article on the fuel oil situation written by Kalman Steiner and published in the July HPAC is very interesting and seems very timely. The statistics I have been able to check seem to be reasonably accurate and the conclusions as applying to the country as a whole seem unquestionably in order.

As Mr. Steiner points out, the threatened shortage of petroleum products applies almost exclusively to the Atlantic Seaboard area and is due not to a lack of production capacity but to a shortage of transportation facilities. Mr. Steiner states that this shortage was caused by transferring 50 tankers out of a total of 325 tankers serving this area to Britain. This is approximately 15 per cent of the number of tankers in this service but it represents considerably more than 15 per cent in transportation volume, I believe, since among the transferred tankers were some of the largest capacity ships of over 15 knots speed. Although a number of shipyards are now constructing tankers, few of

"Open for Discussion"

these will be completed before 1942 and there is no assurance even then that they will be placed in the Atlantic Coast trade. The time element also applies to the construction of pipe lines, some of which are already partially completed but which cannot be placed in service until some time next year. If additional tankers are withdrawn from the Atlantic Coast trade, it may well precipitate a desperate situation.

It therefore appears to me that, although there may not be a shortage over a term of years, there is a distinct possibility of an acute shortage on the Atlantic Seaboard during the winter of 1941-42. This feeling seems to be shared by the companies distributing fuel oil in New York City, some of whom report that they are being limited by their sources of supply to 80 per cent of their volume of sales for the past 12 months. If such a reduction in commitments becomes general, it will mean either that fuel oil will be delivered to preferred customers or else that consumers will eventually be rationed. Should such a rationing occur, based on a percentage of last year's consumption, it would place a distinct penalty upon plants which were operated at good efficiency and in which reductions in consumption could not be made without sacrificing service. Plants which were operated inefficiently would be able to make adjustments and install controls to improve operating efficiency and provide good service with the rationed quantity of oil. Therefore it is to be hoped that, should rationing occur, some yardstick of efficiency, such as the consumption per cubic foot per degree day, will be applied.

It has been possible for a number of years to obtain fuel oil contracts based upon the posted harbor price plus a stated delivery charge, with the stipulation that the price would not exceed a set amount. This year contracts are offered to supply requirements at posted prices with no stipulated maximum. This adds to the uncertainty of the operator.

Mr. Steiner refers in his article to the fact that fuel oil is essentially a byproduct of gasoline manufacture and that gasoline production will be stepped up to meet defense requirements. However, the percentage of fuel oil obtained from crudes can be controlled to a considerable extent in some of the new refining methods, which leave very small percentages of residue which can be used as fuel oil.

As a result of all of these factors, a number of plant owners have made studies of the possibility of converting boilers to burn coal instead of fuel oil and, in some cases, grates or stokers have been installed, even though an increased operating cost may be indicated. In other cases, preparations are being made to purchase grate bars and new fronts to be installed only if fuel oil becomes unavailable. In some plants having multiple boilers, one or more may be converted to burn solid fuel in the belief that there will always be some fuel oil available and that burning of some coal, at least on the day shifts, will indicate a willingness on the part of the operator to coöperate in reducing petroleum consumption and should induce the suppliers to provide sufficient oil for the oil-burning boilers.

Neither government agencies nor oil companies can give any guarantee as to whether or not sufficient fuel oil to meet all requirements will be available in this area, so that the cautious plant operator feels obligated to be in a position to provide heat and power from a secondary source in case of failure of the primary source.—L. W. Mauger, Brown, Wheelock, Harris, Stevens, Inc., (real estate management).

Mr. Steiner Replies

The Editor—

Mr. Mauger makes some very thoughtful comments on the emergency fuel oil situation.

The 50 tankers given to Britain might actually represent somewhat more than 15 per cent of the capacity of the 325 originally serving the East Coast by virtue of the larger and speedier craft being included in the 50. But the capacity of the remaining 275 has been increased by a ruling permitting heavier loading than formerly.

Should an actual shortage develop during the winter, and rationing become necessary, might it not seem more logical to ration gasoline used in pleasure driving, rather than the fuel required to heat homes and buildings? Curtailment of pleasure driving would accomplish other desirable economies from the viewpoint of defense requirements, as for instance rubber and lubricating oils and greases. Furthermore, since a 50 per cent curtailment in automobile production has been ordered it would be well to start thinking of reducing automobile wear and tear.

Mr. Mauger raises the point that rationing of fuel oil might penalize those plants that are already operating at better than average efficiency, and suggests that rationing be based upon some such yardstick as consumption of heating fuel per cubic foot per degree day. The thought is excellent, but who will establish the standards of comparison? A great many factors are involved, and unnecessary hardship might be imposed.

Mr. Mauger states that certain refiners are reluctant to make contracts with maximum prices on fuel oil for the coming season. While in previous years it has been the practice to establish such maximums, it would probably be found upon studying the records that the actual market price during the season seldom reached the established maximum.* In fact, many middlewestern fuel oil distributors have found over the years that a large number of their customers prefer to buy fuel oil at open market prices.

Much progress has been made in recent years in the development of so-called catalytic refining processes, whereby the gasoline yield is greatly increased at the expense of the fuel oil fractions, but these processes are by no means perfected or in general use. By contrast it might be pointed out that whereas the cracking still is the accepted process for major refineries, a great many independent refiners employ only a topping still, thereby failing to take advantage of the refining technology developed during the past 30 years.—Kalman Steiner, Ace Engineering Co.

*Mr. Mauger does not agree, and cites examples.—Ed.

Intricate Piping Is Needed for Fluorescent Lamp Manufacture

O. R. WOLLENTIN EXPLAINS PIPING SERVICES AT NEW WESTINGHOUSE WINDOWLESS FACTORY

PIPING services required in a fluorescent lamp manufacturing plant include many types of gases, in addition to the water piping for process work and cooling purposes, heating lines, sewer systems, etc. In a lamp making plant, a quality product requires quality services and to obtain such conditions special control systems are used.

An excellent example of the importance of piping in this industry is given by the new Westinghouse plant near Fairmont, W. Va.

Natural Gas

Natural gas is used for heating the building and for burners on lamp making machinery. Individual gas fired units were chosen for heating, some being connected to a duct system supplying the various offices, cafeteria, recreation rooms and bulb coating room through air diffusers situated in the ceilings of such rooms. All of the units are automatically controlled by temperature devices so that the temperature of the plant can be held constant.

By this arrangement, flexibility of heat control can be obtained so that in spots where local heat is being given off by lamp machinery heating units need only be used during starting up periods until the machinery is operating; they can then be shut off entirely or set to some lower temperature as required. This should result in efficiency both from the maintenance and fuel economy viewpoints.

In the summer months, the same ducts are used for cooling the various locations by passing water through cooling coils installed in the ducts. Water for cooling purposes is recirculated through a heat exchanger cooled by mine water obtained at approximately 55 F.

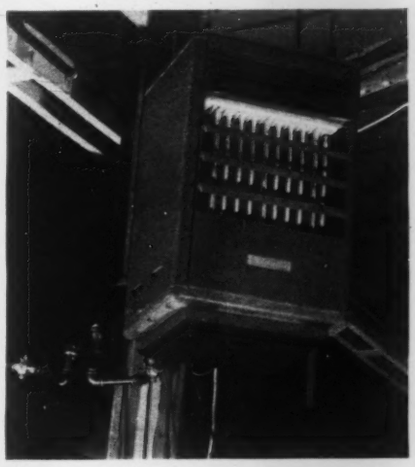
The gas supply to the heating units is from a line independent of the line for the lamp machinery. The gas line supplying the lamp machinery is in the form of a loop so that pressure regulation due to line drop is held to a minimum. Short branch lines from this loop supply the individual machines and where extremely fine regulation is required, the branch lines are also in the form of a loop system. The entire loop system is maintained at a predetermined constant pressure by commercial regulators and recording instruments provide a daily record of operating conditions.

Compressed Air

Gas fires on lamp machinery require compressed air at approximately 4 psi pressure for best operation. This air is supplied by standard centrifugal air compressors taking air from inside the building (since the air in the building is all filtered through electrostatic air cleaners) and is piped in a manner similar to the loop system used for the gas lines. The gas and air lines are practically identical systems. The air lines to each machine are also guarded by a special air line filter to eliminate any possible foreign material getting into the burners, as the burners have very small openings which could be easily clogged up by any dirt.

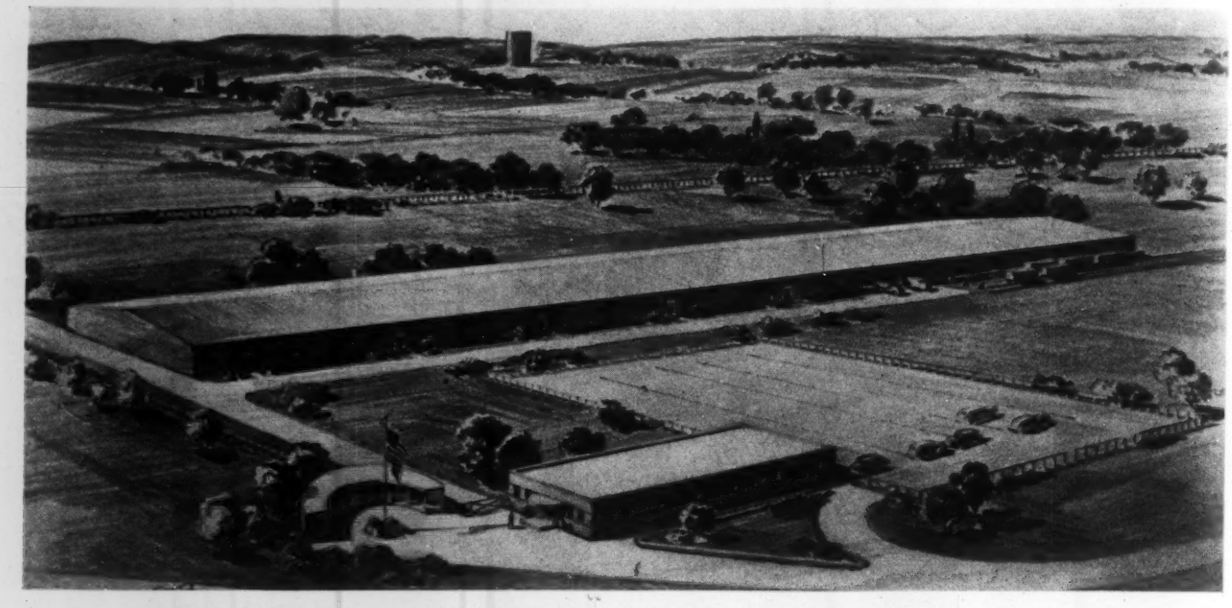
Oxygen

On some lamp machinery, oxygen with gas or air is required. The oxygen cascade system consists of a number of cylinders housed in a separate building approximately 350 ft away because of the hazards encountered in handling oxygen. The cylinders contain oxygen gas at 2000 psi pressure connected to a manifold system and reduced to approximately 100 psi pressure which is piped underground into the main building where it is immediately reduced to a lower pressure (10 psi approximately) before being distributed throughout the plant. All piping and fittings for oxygen gas are made of copper to eliminate corrosion, and at the same time to reduce the hazards possible in oxygen



Thermostatically controlled gas fired unit heaters range in capacities from 85,000 to 200,000 Btu per hr with air deliveries ranging from 1353 to 2260 cfm

INTRICATE PIPING—Piping services in a fluorescent lamp manufacturing plant—such as the new Westinghouse factory near Fairmont, W. Va.—include many types of gases, water for cooling and process, heating lines, gas lines, etc. Mr. Wollentin, equipment superintendent of the Westinghouse lamp division, describes the piping details and other interesting things about this new windowless, air conditioned plant. . . . Gas fired unit heaters supplied with natural gas are used for heating. Air conditioning is accomplished by means of 55 F water pumped from an abandoned coal mine cooling city water which is circulated through the air conditioning coils. All air is cleaned electrostatically, as dust free air is essential to the manufacturing process. And to make the plant a complete "controlled conditions" enclosure for the health and efficiency of the personnel, sterilizing lamps are used



KEEP IT CLEAN was the rule—for dust free air is particularly important in fluorescent lamp manufacture; dust particles clinging to the inside of the glass tubes before coating produce irregularities that may affect operation of the finished lamp. . . . The building is therefore windowless, with all air passed through electrostatic air cleaning cells. . . . Because it is windowless, air conditioning was required . . . . . Fluorescent lighting is, appropriately, used throughout

The new Westinghouse fluorescent lamp plant near Fairmont, W. Va., is a $3,000,000 windowless, zone air conditioned manufacturing plant with nearly five acres of floor space under one roof, and an ultimate output of 200,000 lamps a day. It is 100 per cent fluorescently lighted. Auxiliary buildings, paved roads and driveways, railroad sidings, water reservoir, bus terminal, parking lot and other incidental structures, all enclosed by fence and floodlighted for protection against sabotage, will complete the project, as shown in the above architect's drawing.

To meet the demand for fluorescent lamps—accelerated by the increasing number of "blackout" industrial plants built during the last few months—an extensive increase in lamp production capacity was required. Thus the factory might be considered a "secondary" defense plant. One of the general requirements for the plant site was an adequate supply of fuel gas, both for the lamp manufacturing plant and for a projected glass factory. Transportation facilities, electric power, water service and an adequate supply of labor suited to lamp manufacture were other factors to be considered.

Windowless design, with its necessity for complete air conditioning, was not adopted with the intention of providing a blackout plant in the current sense, but was necessary because working conditions must be closely controlled. Dust free air is particularly important in manufacture of fluorescent lamps; dust particles clinging to the inside of the glass tubes before coating produce irregularities that may affect operation of the finished lamp. For this reason the air conditioning system incorporates electrostatic air cleaning cells, which remove approximately 90 per cent of all air-borne particles before incoming air is circulated in the plant. Air in all ducts is irradiated by a total of 600 sterilizing lamps to destroy bacteria and other microörganisms. Shipping and receiving rooms are provided with "air curtains" (separated double doors with air chambers between) to help keep out dust.

The air conditioning system is provided with zone control to keep temperature fairly uniform throughout the plant, despite widely varying processing conditions, and (secondarily) to make sabotage or bombardment less effective, if that ever becomes especially important. Because some 40 million cubic feet of natural gas per month will be burned in glass working, air is not recirculated in the main manufacturing area. Proportion of outside air to recirculated air in offices may be adjusted to suit requirements. Air ducts are supported on the lower chords of trusses extending from the coils and blowers located in the same general position. When heat is required in the building, gas fired unit heaters are used.

An unusual part of the air conditioning system is the method of air cooling. It comprises two separate systems of circulating water. The primary system consists of a 3600 gpm flow of water at 55 F pumped from an abandoned and sealed coal mine 1800 ft away and 150 ft lower in elevation. Mine water is pumped to a heat exchanging reservoir just outside the building, thence back to the coal mine through a drill hole. The secondary system consists of a closed circulation of ordinary city water, through cooling coils in the building, thence to the heat exchanging reservoir, where heat is extracted and carried away in the primary system. Heat exchange is accomplished by copper coils completely submerged in the mine water.

gas lines. Special pipe line filters at each machine are used to eliminate foreign material.

The cylinders are refilled from a special oxygen service truck which delivers oxygen in liquid form and by means of special devices on the truck converting the liquid to gas and pumping it into the cylinders to approximately 2000 psi pressure.

Argon and Hydrogen

Argon gas and hydrogen gas are also housed in a separate building approximately 350 ft from the main building and piped through an underground conduit into the main building. Both of these gases are obtained in cylinders at high pressure and are housed in the separate buildings to avoid handling cylinders in the manufacturing section.

The hydrogen system consists of two banks of cylinders, one operating and one for servicing or standby duty, both feeding into duplicate high pressure pipe lines through regulators and reduced to approximately 100 psi pressure before being piped underground to the main building. Here it is again reduced to approximately 10 psi before being distributed through the factory.

The argon system consists of two banks of cylinders, each connected to its own high pressure (approximately 2600 psi) copper line, both of which are carried through the underground conduit to a high pressure reducing panel in the manufacturing area. At the panel, the pressure is reduced to approximately 15 psi and distributed to the machines where the gas is again controlled to the proper lamp filling pressure. The control panel consists of high pressure reducing regulators, gages, alarm systems for low pressure, cross connecting piping and valves so that either bank of cylinders can be used to supply the machines.

Check Line Tests Quality

A quality check line is run to a special machine so that when gas cylinders are changed, proper tests can be made at one machine without contaminating other machines and lines in case the gas should not measure up to the proper standard.

The argon lines were specially treated before installation to avoid any possibility of contamination from dirt, oil and moisture. When the manifolds and lines had been fabricated in the mechanical departments (machine shop, welder, pipe fitting), they were sent through a cleaning and annealing process which consists of flushing the parts with steam. They were then put into a furnace and brought up to a temperature of approximately 400 C, with nitrogen constantly flowing through the copper tubing and manifolds. This drove out all impurities in the parts, and after taking them from the furnace, the parts were capped and sealed to avoid any chance of contamination.

Since all the parts were prefabricated to proper sizes, the only installation required was placing in the proper conduit or trough and assembly. No cutting or fitting was done in the field, all parts being made in our shops. When a line was completed, it was first filled with nitrogen and tests made for leaks. When tight, the line was flushed with argon until proper quality was obtained and the line was ready for use.

Mine Water for Cooling

The process water supply for air cooling and machine cooling was somewhat of a problem because the location of the plant is such that well water was not available in the quantities desired, and water from the nearby Monongahela River was not suitable for our purpose.

A shot of some of the welded piping taken during the construction period

The only other source is underground water stored up in abandoned and outworked coal mines. This is cold enough, approximately 55 F, and the supply is adequate, but there was the problem of acidity, which was high. After considering all other sources, it was decided to use the mine water for air cooling purposes and city water for all other purposes. The mine water is pumped into a tank in which there are coils of special metal. City water is circulated through these coils and cooled to within 2 or 3 F of the mine water temperature. The cold city water is then circulated through coils in the air ducts throughout the factory and is then recirculated through the mine water tank for cooling.

All outside air taken into the building is passed through electrostatic air cleaners, thereby insuring cleanliness, which is a very important feature in the making of fluorescent lamps. All air ducts are equipped with sterilizing lamps to increase the efficiency of the plant both from a personnel and quality product standpoint.

Two 50 psi air compressors, with aftercoolers and oil separators, furnish air for various sundry purposes such as air motors, cleaning operations and some special movements on lamp making machines. The sewer and plumbing systems are conventional except for modifications to meet unexpected rock obstructions.

RADIANT HEATING AND COOLING

In the third paragraph of the article on radiant heating and cooling by F. E. Giesecke, p. 589, October, 1940, HPAC, a calculation was shown which was in error. The correct solution is as follows:

The volume of air moving across the ceiling in one hour is $12 \times 0.75 \times 50 \times 60$ or 27,000 cu ft per hr, assuming that the registers are at the narrow ends of the room. The weight of this air is 1850 lb. It will deliver 463 Btu per deg temperature drop. To deliver 6000 Btu requires a drop in temperature of 13 deg.

Hence the air should enter the room at 126 F and leave at 113, instead of entering at 124 F and leaving at 114.

CHILLED WATER STORAGE TANKS ALLOW ECONOMIES IN COMPRESSOR OPERATION

New Data for Quickly Sizing Tanks for Air Conditioning and Refrigerating Systems Presented by William Goodman

STORAGE TANKS — Wherever chilled water or brine is circulated for air conditioning or refrigerating purposes, the use of a storage tank permits operating economies to be achieved. By using a storage tank, the compressor can be operated at practically full load whenever it is running. Under light load, the running time of the compressor will be about directly proportional to the cooling load with resultant economy in electricity and condensing water. Furthermore, the use of a storage tank prevents short cycling of the compressor under light load. . . . The method of properly sizing storage tanks so that they are neither too large nor too small is discussed in a practical, how-to-do-it manner in this article and tables for quickly computing the required size of storage tanks are presented. The data given are unavailable elsewhere and have been developed by Bill Goodman, consulting engineer, The Trane Co., and member of HPAC's board of consulting and contributing editors, for presentation here

STORAGE tanks containing chilled water are frequently of value in a refrigerating system in which the air is cooled by circulating chilled water. Storage tanks can provide marked economies in the electrical consumption of the compressor, and in addition prevent the frequent cycling—that is, the frequent starting and stopping—of the compressor with its attendant wear and tear. In low temperature refrigeration work, brine is, of course, circulated, and the information given here in regard to chilled water circulating systems is equally applicable to brine circulating systems.

When the refrigerating load is light, a reciprocating compressor draws almost as much power as at full load. A storage tank used in conjunction with a compressor will allow it to operate at just about full load whenever it is running. The time that the compressor runs will be practically in direct proportion to the cooling load. Thus, if the cooling load is one-half the maximum, the compressor will run just about half the time cooling is required if an insulated storage tank is used. In this way the electrical energy used by the compressor will be almost directly in proportion to the load and, consequently, substantial savings in power can frequently be effected by the use of a storage tank.

Under light load, the use of a water storage tank prevents frequent cycling of the compressor. The storage tank can be so sized that when the compressor does operate, it will run for not less than a definite time.

Symbols

c = specific heat of stored liquid;
H_c = refrigerating capacity of the compressor, Btu per min;
H_L = cooling load, Btu per min;
H_s = thermal energy stored in tank, Btu;
s = specific gravity of stored liquid;
t_1 = initial temperature of liquid in storage tank, F;
t_2 = final temperature of liquid in storage tank, F;
θ = total time of cycle = sum of on and off times of compressor, min;
θ_c = time compressor runs, min.

Sizing the Storage Tank

Table 1 can be used for conveniently and quickly determining the size of a storage tank required for systems in which the installed *compressor capacity* is equal to the *maximum cooling load*. During periods of light cooling load, the compressor capacity will obviously be *greater* than the cooling load. The storage tanks discussed in this article are of the type in which the stored fluid remains liquid; no freezing occurs.

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The figures across the top of Table 1 show the ratio of the actual cooling load to the compressor capacity—that is, the load ratio. In Table 1, load ratios up to 0.50 are listed at the top of the various columns; load ratios greater than 0.50 are listed at the bottom of the columns.

The figures in the body of Table 1 give the length of time that the compressor will operate, the length of time that it will be off, and the total time of the cycle—which is the sum of the "on" and "off" times—for the various storage capacities listed in columns 1 and 32.

The figures in columns 1 and 32 (which are headed "Storage Capacity") represent the storage capacity in Btu per ton of compressor capacity. To determine the volume of water or brine to be stored, the following formula may be used:

$$\text{Gallons} = \frac{H_s}{8.33\, sc\,(t_2 - t_1)} \quad \ldots\ldots [1]$$

For convenience, storage data for water are presented in Table 2. The data in this table have been prepared by means of Formula 1. To use Table 2 for brine or any fluid other than water, divide the "gallons of water storage" by the product of the specific gravity and the specific heat of the fluid being used.

When using either Formula 1 or Table 2, the allowable temperature rise of the water in the storage tank must first be determined. This rise is determined by the maximum temperature at which water may be supplied to the air cooling equipment and the lowest suction temperature at which the compressor is to be allowed to operate. The rise in temperature of the water in the storage tank need not necessarily be the same as the rise in temperature of the water circulating through the air cooling coil. Indeed, the two temperature rises are generally dif-

Table 1 On and off times of compressors with storage systems in which compressor capacity is *greater* than the average load. (On, off, and cycle times are in hours and minutes. Storage capacity is given in Btu per ton of compressor capacity.)

Table 1. **On *and off times* of compressors with storage systems in which compressor capacity is *greater* than the average load. (On, off, and cycle times are in hours and minutes. Storage capacity is given in Btu per ton of compressor capacity)**

Storage Capacity	Load / Compressor Capacity																																	Storage Capacity
	0.05			0.10			0.15			0.20			0.25			0.30			0.35			0.40			0.45			0.50					
	On	Off	Cycle	On	Off	Cycle	On	Off	Cycle	On	Off	Cycle	On	Off	Cycle	On	Off	Cycle	On	Off	Cycle	On	Off	Cycle	On	Off	Cycle	On	Off	Cycle			
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32			
500	0- 3'	0-49'	0-52'	0- 3'	0-25'	0-28'	0- 3'	0-16'	0-19'	0- 3'	0-13'	0-16'	0- 3'	0-10'	0-13'	0- 4'	0- 8'	0-12'	0- 4'	0- 7'	0-11'	0- 4'	0- 6'	0-10'	0- 4'	0- 6'	0-10'	0- 5'	0- 5'	0-10'	500			
1000	0- 5	1-41	1-46	0- 6	0-50	0-56	0- 6	0-33	0-39	0- 6	0-25	0-31	0- 7	0-20	0-27	0- 7	0-37	0-44	0- 8	0-14	0-22	0- 8	0-13	0-21	0- 9	0-11	0-20	0-10	0-10	0-20	1000			
2000	0-11	3-20	3-31	0-11	1-40	1-51	0-12	1- 7	1-19	0-12	0-50	1- 2	0-13	0-40	0-53	0-14	0-33	0-47	0-15	0-29	0-44	0-17	0-25	0-42	0-18	0-22	0-40	0-20	0-20	0-40	2000			
3000	0-16	5- 0	5-16	0-17	2-30	2-47	0-18	1-40	1-58	0-19	1-15	1-34	0-20	1- 0	1-20	0-21	0-50	1-11	0-23	0-43	1- 6	0-25	0-38	1- 3	0-27	0-33	1- 0	0-30	0-30	1- 0	3000			
4000	0-21	6-41	7- 2	0-22	3-20	3-42	0-24	2-13	2-37	0-25	1-40	2- 5	0-27	1-20	1-47	0-29	1- 7	1-36	0-31	0-57	1-28	0-33	0-50	1-23	0-36	0-44	1-20	0-40	0-40	1-20	4000			
5000	0-26'	8-20'	8-46'	0-28'	4-10'	4-38'	0-29'	2-47'	3-16'	0-31'	2- 5'	2-36'	0-33'	1-40'	2-13'	0-36'	1-23'	1-59'	0-38'	1-12'	1-50'	0-42'	1- 2'	1-44'	0-45'	0-56'	1-41'	0-50'	0-50'	1-40'	5000			
6000	0-32	10- 0	10-32	0-33	5- 0	5-33	0-35	3-20	3-55	0-38	2-30	3- 8	0-40	2- 0	2-40	0-43	1-40	2-23	0-46	1-26	2-12	0-50	1-15	2- 5	0-55	1- 6	2- 1	1- 0	1- 0	2- 0	6000			
7000	0-37	11-39	12-16	0-39	5-50	6-29	0-41	3-54	4-35	0-44	2-55	3-39	0-47	2-20	3- 7	0-50	1-56	2-46	0-54	1-40	2-34	0-58	1-28	2-26	1- 4	1-17	2-21	1-10	1-10	2-20	7000			
8000	0-42	13-20	14- 2	0-44	6-40	7-24	0-47	4-27	5-14	0-50	3-20	4-10	0-53	2-40	3-33	0-57	2-13	3-10	1- 2	1-54	2-56	1- 7	1-40	2-47	1-13	1-28	2-41	1-20	1-20	2-40	8000			
9000	0-47	14-59	15-46	0-50	7-30	8-20	0-53	5- 0	5-53	0-56	3-45	4-41	1- 0	3- 0	4- 0	1- 4	2- 0	3-34	1- 9	2- 9	3-18	1-15	1-52	3- 7	1-22	1-40	3- 2	1-30	1-30	3- 0	9000			
10,000	0-53'	16-39'	17-32'	0-56'	8-20'	9-16'	0-59'	5-33'	6-32'	1- 2'	4-10'	5-12'	1- 7'	3-20'	4-27'	1-11'	2-46'	3-57'	1-17'	2-23'	3-40'	1-23'	2- 5'	3-28'	1-31'	1-51'	3-22'	1-40'	1-40'	3-20'	10,000			
11,000	0-58	18-20	19-18	1- 1	9-10	10-11	1- 5	6- 7	7-12	1- 9	4-35	5-44	1-13	3-40	4-53	1-19	3- 3	4-22	1-25	2-37	4- 2	1-32	2-17	3-49	1-40	2- 2	3-42	1-50	1-50	3-40	11,000			
12,000	1- 3	20- 1	21- 4	1- 7	10- 0	11- 7	1-11	6-40	7-51	1-15	5- 0	6-15	1-20	4- 0	5-20	1-26	3-20	4-46	1-32	2-52	4-24	1-40	2-30	4-10	1-49	2-13	4- 2	2- 0	2- 0	4- 0	12,000			
13,000	1- 8	21-40	22-48	1-12	10-50	12- 2	1-16	7-14	8-30	1-21	5-25	6-46	1-27	4-20	5-47	1-33	3-36	5- 9	1-40	3- 6	4-46	1-48	2-43	4-31	1-58	2-24	4-22	2-10	2-10	4-20	13,000			
14,000				1-18	11-40	12-58	1-22	7-47	9- 9	1-28	5-50	7-18	1-33	4-40	6-13	1-40	3-53	5-33	1-48	3-20	5- 8	1-57	2-55	4-52	2- 7	2- 9	3-18	2-35	4-42	2-20	2-20	4-40	14,000			
15,000				1-23'	12-30'	13-53'	1-28'	8-20'	9-48'	1-34'	6-15'	7-49'	1-40'	5- 0'	6-40'	1-47'	4-10'	5-57'	1-55'	3-35'	5-30'	2- 5'	3- 8'	5-13'	2-16'	2-46'	5- 2'	2-30'	2-30'	5- 0'	15,000						
16,000				1-29	13-20	14-49	1-34	8-54	10-28	1-40	6-40	8-20	1-47	5-20	7- 7	1-54	4-26	6-20	2- 3	3-49	5-52	2-13	3-20	5-33	2-25	2-58	5-23	2-40	2-40	5-20	16,000						
17,000				1-34	14-10	15-44	1-40	9-27	11- 7	1-46	7- 5	8-51	1-53	5-40	7-33	2- 1	4-43	6-44	2-11	4- 3	6-14	2-22	3-32	5-54	2-35	3- 8	5-43	2-50	2-50	5-40	17,000						
18,000				1-40	15- 0	16-40	1-46	10- 0	11-46	1-52	7-30	9-22	2- 0	6- 0	8- 0	2- 9	5- 0	7- 9	2-18	4-18	6-36	2-30	3-45	6-15	2-44	3-19	6- 3	3- 0	3- 0	6- 0	18,000						
19,000				1-46	15-50	17-36	1-52	10-34	12-26	1-59	7-55	9-54	2- 7	6-20	8-27	2-16	5-16	7-32	2-26	4-32	6-58	2-38	3-58	6-36	2-53	3-30	6-23	3-10	3-10	6-20	19,000						
20,000				1-51'	16-40'	18-31'	1-58'	11- 7'	13- 5'	2- 5'	8-20'	10-25'	2-13'	6-40'	8-53'	2-23'	5-33'	7-56'	2-34'	4-46'	7-20'	2-47'	4-10'	6-57'	3- 2'	3-42'	6-44'	3-20'	3-20'	6-40'	20,000						
21,000				1-57	17-30	19-27	2- 4	11-40	13-44	2-11	8-45	10-56	2-20	7- 0	9-20	2-30	5-50	8-20	2-42	5- 0	7-42	2-55	4-22	7-17	3-11	3-53	7- 4	3-30	3-30	7- 0	21,000						
22,000				2- 2	18-20	20-22	2- 9	12-14	14-23	2-18	9-10	11-28	2-27	7-20	9-47	2-37	6- 6	8-43	2-49	5-15	8- 4	3- 3	4-35	7-38	3-20	4- 4	7-24	3-40	3-40	7-20	22,000						
23,000				2- 8	19-10	21-18	2-15	12-47	15- 2	2-24	9-35	11-59	2-33	7-40	10-13	2-44	6-23	9- 7	2-57	5-29	8-26	3-12	4-48	8- 0	3-29	4-15	7-44	3-50	3-50	7-40	23,000						
24,000				2-13	20- 0	22-13	2-21	13-21	15-42	2-30	10- 0	12-30	2-40	8- 0	10-40	2-51	6-39	9-30	3- 5	5-43	8-48	3-20	5- 0	8-20	3-38	4-26	8- 4	4- 0	4- 0	8- 0	24,000						
25,000				2-19'	20-50'	23- 9'	2-27'	13-54'	16-21'	2-36'	10-25'	13- 1'	2-47'	8-20'	11- 7'	2-59'	6-56'	9-55'	3-12'	5-58'	9-10'	3-28'	5-13'	8-41'	3-47'	4-37'	8-24'	4-10'	4-10'	8-20'	25,000						
26,000							2-33	14-27	17- 0	2-42	10-50	13-32	2-53	8-40	11-33	3- 6	7-13	10-19	3-20	6-12	9-32	3-37	5-25	9- 2	3-56	4-49	8-45	4-20	4-20	8-40	26,000			
27,000							2-39	15- 0	17-39	2-49	11-15	14- 4	3- 0	9- 0	12- 0	3-13	7-29	10-42	3-28	6-26	9-54	3-45	5-37	9-22	4- 5	5- 0	9- 5	4-30	4-30	9- 0	27,000			
28,000							2-45	15-34	18-19	2-55	11-40	14-35	3- 7	9-20	12-27	3-20	7-46	11- 6	3-35	6-41	10-16	3-53	5-50	9-43	4-15	5-10	9-25	4-40	4-40	9-20	28,000			
29,000							2-51	16- 7	18-58	3- 1	12- 5	15- 6	3-13	9-40	12-53	3-27	8- 3	11-30	3-43	6-55	10-38	4- 2	6- 3	10- 5	4-24	5-21	9-45	4-50	4-50	9-40	29,000			
30,000							2-56'	16-41'	19-37'	3- 8'	12-30'	15-38'	3-20'	10- 0'	13-20'	3-34'	8-19'	11-53'	3-51'	7- 9'	11- 0'	4-10'	6-15'	10-25'	4-33'	5-33'	10- 6'	5- 0'	5- 0'	10- 0'	30,000			
31,000							3- 2	17-14	20-16	3-14	12-55	16- 9	3-27	10-20	13-47	3-41	8-36	12-17	3-58	7-24	11-22	4-18	6-27	10-45	4-42	5-44	10-26	5-10	5-10	10-20	31,000			
32,000							3- 8	17-47	20-55	3-20	13-20	16-40	3-33	10-40	14-13	3-49	8-53	12-42	4- 6	7-38	11-44	4-27	6-40	11- 7	4-51	5-55	10-46	5-20	5-20	10-40	32,000			
33,000							3-14	18-21	21-35	3-26	13-45	17-11	3-40	11- 0	14-40	3-56	9- 9	13- 5	4-14	7-52	12- 6	4-35	6-52	11-27	5- 0	5- 6	11- 6	5-30	5-30	11- 0	33,000			
34,000							3-20	18-54	22-14	3-32	14-10	17-42	3-47	11-20	15- 7	4- 3	9-26	13-29	4-22	8- 6	12-28	4-43	7- 5	11-48	5- 9	6-17	11-26	5-40	5-40	11-20	34,000			
35,000							3-26'	19-27'	22-53	3-39'	14-35'	18-14'	3-53'	11-40'	15-33'	4-10'	9-42'	13-52'	4-29'	8-21'	12-50'	4-52'	7-18'	12-10'	5-18'	6-28'	11-46'	5-50'	5-50'	11-40'	35,000			
36,000							3-32	20- 1	23-33	3-45	15- 0	18-45	4- 0	12- 0	16- 0	4-17	9-59	14-16	4-37	8-35	13-12	5- 0	7-30	12-30	5-27	6-39	12- 6	6- 0	6- 0	12- 0	36,000			
37,000										3-51	15-25	19-16	4- 7	12-20	16-27	4-24	10-16	14-40	4-45	8-49	13-34	5- 8	7-42	12-50	5-36	6-51	12-27	6-10	6-10	12-20	37,000			
38,000										3-58	15-50	19-48	4-13	12-40	16-53	4-31	10-32	15- 3	4-52	9- 4	13-56	5-17	7-55	13-12	5-45	7- 2	12-47	6-20	6-20	12-40	38,000			
39,000										4- 4	16-15	20-19	4-20	13- 0	17-20	4-39	10-49	15-28	5- 0	9-18	14-18	5-25	8- 8	13-33	5-55	7-12	13- 7	6-30	6-30	13- 0	39,000			
40,000										4-10'	16-40'	20-50'	4-27'	13-20'	17-47'	4-46'	11- 6'	15-52'	5- 8'	9-32'	14-40'	5-33'	8-20'	13-53'	6- 4'	7-23'	13-27'	6-40'	6-40'	13-20'	40,000			
41,000										4-16	17- 5	21-21	4-33	13-40	18-13	4-53	11-22	16-15	5-15	9-47	15- 2	5-42	8-32	14-14	6-13	7-34	13-47	6-50	6-50	13-40	41,000			
42,000										4-22	17-30	21-52	4-40	14- 0	18-40	5- 0	11-39	16-39	5-23	10- 1	15-24	5-50	8-45	14-35	6-22	7-46	14- 8	7- 0	7- 0	14- 0	42,000			
43,000										4-29	17-55	22-24	4-47	14-20	19- 7	5- 7	11-56	17- 3	5-31	10-15	15-46	5-58	8-57	14-55	6-31	7-57	14-28	7-10	7-10	14-20	43,000			
44,000										4-35	18-20	22-55	4-53	14-40	19-33	5-14	12-12	17-26	5-38	10-30	16- 8	6- 7	9-10	15-17	6-40	8- 8	14-48	7-20	7-20	14-40	44,000			
45,000										4-41'	18-45'	23-26'	5- 0'	15- 0'	20- 0'	5-21'	12-29'	17-50'	5-46'	10-44'	16-30'	6-15'	9-22'	15-37'	6-49'	8-19'	15- 8'	7-30'	7-30'	15- 0'	45,000			
46,000										4-48	19-10	23-58	5- 7	15-20	20-27	5-29	12-46	18-15	5-54	10-58	16-52	6-23	9-35	15-58	6-58	8-30	15-28	7-40	7-40	15-20	46,000			
47,000													5-13	15-40	20-53	5-36	13- 2	18-38	6- 2	11-12	17-14	6-32	9-48	16-20	7- 7	8-41	15-48	7-50	7-50	15-40	47,000			
48,000													5-20	16- 0	21-20	5-43	13-19	19- 2	6- 9	11-27	17-36	6-40	10- 0	16-40	7-16	8-53	16- 9	8- 0	8- 0	16- 0	48,000			
49,000													5-27	16-20	21-47	5-50	13-36	19-26	6-18	11-41	17-58	6-48	10-12	17- 0	7-25	9- 4	16-29	8-10	8-10	16-20	49,000			
50,000													5-33'	16-40'	22-13'	5-57'	13-52'	19-49'	6-25'	11-55'	18-20'	6-57'	10-25'	17-22'	7-35'	9-14'	16-49'	8-20'	8-20'	16-40'	50,000			
51,000													5-40	17- 0	22-40	6- 4	14- 9	20-13	6-32	12-10	18-42	7- 5	10-38	17-43	7-44	9-25	17- 9	8-30	8-30	17- 0	51,000			
52,000													5-47	17-20	23- 7	6-11	14-25	20-36	6-40	12-24	19- 4	7-13	10-50	18- 3	7-53	9-36	17-29	8-40	8-40	17-20	52,000			
53,000													5-53	17-40	23-33	6-19	14-42	21- 1	6-48	12-38	19-26	7-22	11- 3	18-25	8- 2	9-48	17-50	8-50	8-50	17-40	53,000			
54,000													6- 0	18- 0	24- 0	6-26	14-59	21-25	6-55	12-53	19-48	7-30	11-15	18-45	8-11	9-59	18-10	9- 0	9- 0	18- 0	54,000			
55,000																6-33'	15-15'	21-48'	7- 3'	13- 7'	20-10'	7-38'	11-27'	19- 5'	8-20'	10-10'	18-30'	9-10'	9-10'	18-20'	55,000			
56,000																6-40	15-32	22-12	7-11	13-21	20-32	7-47	11-40	19-27	8-29	10-21	18-50	9-20	9-20	18-40	56,000			
57,000																6-47	15-49	22-36	7-18	13-36	20-54	7-55	11-52	19-47	8-38	10-32	19-10	9-30	9-30	19- 0	57,000			
58,000																6-54	16- 5	22-59	7-26	13-50	21-16	8- 3	12- 5	20- 8	8-47	10-44	19-31	9-40	9-40	19-20	58,000			
59,000																7- 1	16-22	23-23	7-34	14- 4	21-38	8-12	12-18	20-30	8-56	10-55	19-51	9-50	9-50	19-40	59,000			
60,000																7- 9'	16-39'	23-48'	7-42'	14-18'	22- 0'	8-20'	12-30'	20-50'	9- 5'	11- 6'	20-11'	10- 0'	10- 0'	20- 0'	60,000			
61,000																			7-49	14-33	22-22	8-28	12-42	21-10	9-15	11-16	20-31	10-10	10-10	20-20	61,000			
62,000																			7-57	14-47	22-44	8-37	12-55	21-32	9-24	11-27	20-51	10-20	10-20	20-40	62,000			
63,000																			8- 5	15- 1	23- 6	8-45	13- 8	21-53	9-33	11-38	21-11	10-30	10-30	21- 0	63,000			
64,000																			8-12	15-16	23-28	8-53	13-20	22-13	9-42	11-50	21-32	10-40	10-40	21-20	64,000			
65,000																			8-20'	15-30'	23-50'	9- 2'	13-33'	22-35'	9-51'	12- 1'	21-52'	10-50'	10-50'	21-40'	65,000			
66,000																						9-10	13-45	22-55	10- 0	12-12	22-12	11- 0	11- 0	22- 0	66,000			
67,000																						9-18	13-57	23-15	10- 9	12-23	22-32	11-10	11-10	22-20	67,000			
68,000																						9-27	14-10	23-37	10-18	12-34	22-52	11-20	11-20	22-40	68,000			
69,000																						9-35	14-22	23-57	10-27	12-45	23-12	11-30	11-30	23- 0	69,000			
70,000																									10-36'	12-56'	23-32'	11-40'	11-40'	23-20	70,000			
71,000																									10-45	13- 8	23-53	11-50	11-50	23-40	71,000			
72,000																												12- 0	12- 0	24- 0	72,000			
	Off	On	Cycle	Off	On	Cycle	Off	On	Cycle	Off	On	Cycle	Off	On	Cycle	Off	On	Cycle	Off	On	Cycle	Off	On	Cycle	Off	On	Cycle	Off	On	Cycle			
Storage Capacity	0.95			0.90			0.85			0.80			0.75			0.70			0.65			0.60			0.55			0.50			Storage Capacity		
	Load / Compressor Capacity																																	

ferent, for a larger temperature rise is generally allowed in the storage tank.

Because of the change in the temperature of the water in the storage tank, the compressor will operate at different suction temperatures during the period that the water in the tank is cooling. Inasmuch as the capacity of the compressor varies with the suction temperature, it is obvious that the compressor capacity will be changing throughout the cooling period. However, for all practical purposes, the average of the compressor capacities at the initial and final temperatures of the water in the tank is sufficiently accurate, or the capacity at the mean suction temperature.

Example 1 illustrates the use of Tables 1 and 2 for determining the size of storage tank required.

Example 1—Water is to be supplied to a coil cooling air at a temperature not to exceed 50 F. Assume that the water temperature in the tank can be allowed to drop as low as 40 F. Assume also that when the water in the tank is 50 F, the compressor will operate at a suction temperature of 45 F; and when the temperature of the water in the tank is 40 F, the compressor will operate at 35 F. Also assume that at 45 F suction temperature the capacity of the compressor is 24 tons and that at 35 F it is 17 tons. If the minimum load is 35 per cent of the maximum load, find: (a) the storage capacity required if the compressor is to operate for not less than 30 min. (b) the time that the compressor will be off and the total time of a single cycle.

Solution—

a) Average refrigerating capacity = $(24 + 17) \div 2 = 20.5$ tons.

In Column 20 of Table 1, for a load ratio of 0.35, find the nearest running time of 31 min. In either Column 1 or Column 32, in the same row as 31 min, the required storage capacity is found to be 4000 Btu per ton of refrigerating capacity.

Referring to Table 2, for a storage capacity of 4000 Btu and a temperature rise of 10 deg, the storage required is 48 gal per ton of refrigerating capacity. Total storage required is therefore 48 x 20.5 or 984 gal.

b) In Column 21 of Table 1 opposite the running time of 31 min, the off time is found to be 57 min. In Column 22, the total time of the cycle is found to be 1 hr and 28 min.

In Example 1, the value of 984 gal of water represents not only the water stored in the tank, but the water stored in the circulating piping system. However, a margin of safety is obtained by ignoring the water stored in the piping system and making the water storage tank equal to the computed storage capacity of 984 gal.

Once the size of a storage tank has been determined, it is possible to find the running, off and cycle times of the compressor for various ratios of the actual load to the maximum load. The following example illustrates the method of finding the running time for various load percentages.

Example 2—Take the conditions of Example 1 and find the on, off and cycle times for load ratios varying from 0.9 down to 0.1.

Solution—To find the on, off and cycle times for a 0.9 load ratio, with a storage capacity of 4000 Btu, find the load ratio of 0.9 at the bottom of Column 5 in Table 1. In Column 5, opposite a storage capacity of 4000 Btu, the off time is found to be 22 min, and the on time 3 hr and 20 min. In the same way, the values tabulated below (in hours and minutes) were found for all of the various load ratios:

Load Ratio	*On*	*Off*	*Cycle*
0.0	0- 0	Continuously	
0.1	0-22	3-20	3-42
0.2	0-25	1-40	2- 5
0.3	0-29	1- 7	1-36
0.4	0-33	0-50	1-23
0.5	0-40	0-40	1-20
0.6	0-50	0-33	1-23
0.7	1- 7	0-29	1-36
0.8	1-40	0-25	2- 5
0.9	3-20	0-22	3-42
1.0	Continuously	0- 0	

From the data in Example 2, it is obvious that even under very light loads, the storage tank is large enough to keep the compressor running for over 20 min at a time. If a shorter running period is satisfactory, the size of the storage tank may be reduced.

Method of Computing Table 1

When the cooling load is less than the compressor capacity, and a storage tank is used, the running time of the compressor will be less than the time during which cooling is required. The object of the storage tank is to keep the compressor from cycling frequently when the cooling load is less than the maximum.

If the compressor capacity and the minimum cooling load are known, an equation can be developed between the size of storage tank required and the running time of the compressor.

If the compressor operates during the period when the cooling load is less than the compressor capacity, the compressor absorbs heat provided by the cooling load and heat from the water in the storage tank. Hence,

$$H_c\theta_c = H_s + H_L\theta_c \quad \ldots\ldots\ldots [2]$$

$$\theta_c = \frac{H_s}{H_c - H_L} \quad \ldots\ldots\ldots [3]$$

$$\theta_c = \frac{H_s}{H_c\left[1 - \frac{H_L}{H_c}\right]} \quad \ldots\ldots\ldots [4]$$

To find the storage capacity per ton of refrigerating capacity, let $H_c = 200$ Btu per min. Therefore,

$$\theta_c = \frac{H_s}{200\left[1 - \frac{H_L}{H_c}\right]} \quad \ldots\ldots\ldots [5]$$

The compressor running times listed in the on columns of Table 1 were computed by means of Formula 5, for the various values of (H_L/H_c), the load ratio; and H_s, the storage capacity per ton of refrigerating capacity.

Regardless of whether or not the compressor is running, all of the cooling load must eventually be absorbed by the compressor. This is true even though during part of the time the cooling load is carried by the stored thermal energy, because —ultimately—this stored energy must be provided by the compressor. Hence,

$$H_c\theta_c = H_L\theta \quad \ldots\ldots\ldots [6]$$

$$\theta = \theta_c\frac{H_c}{H_L} \quad \ldots\ldots\ldots [7]$$

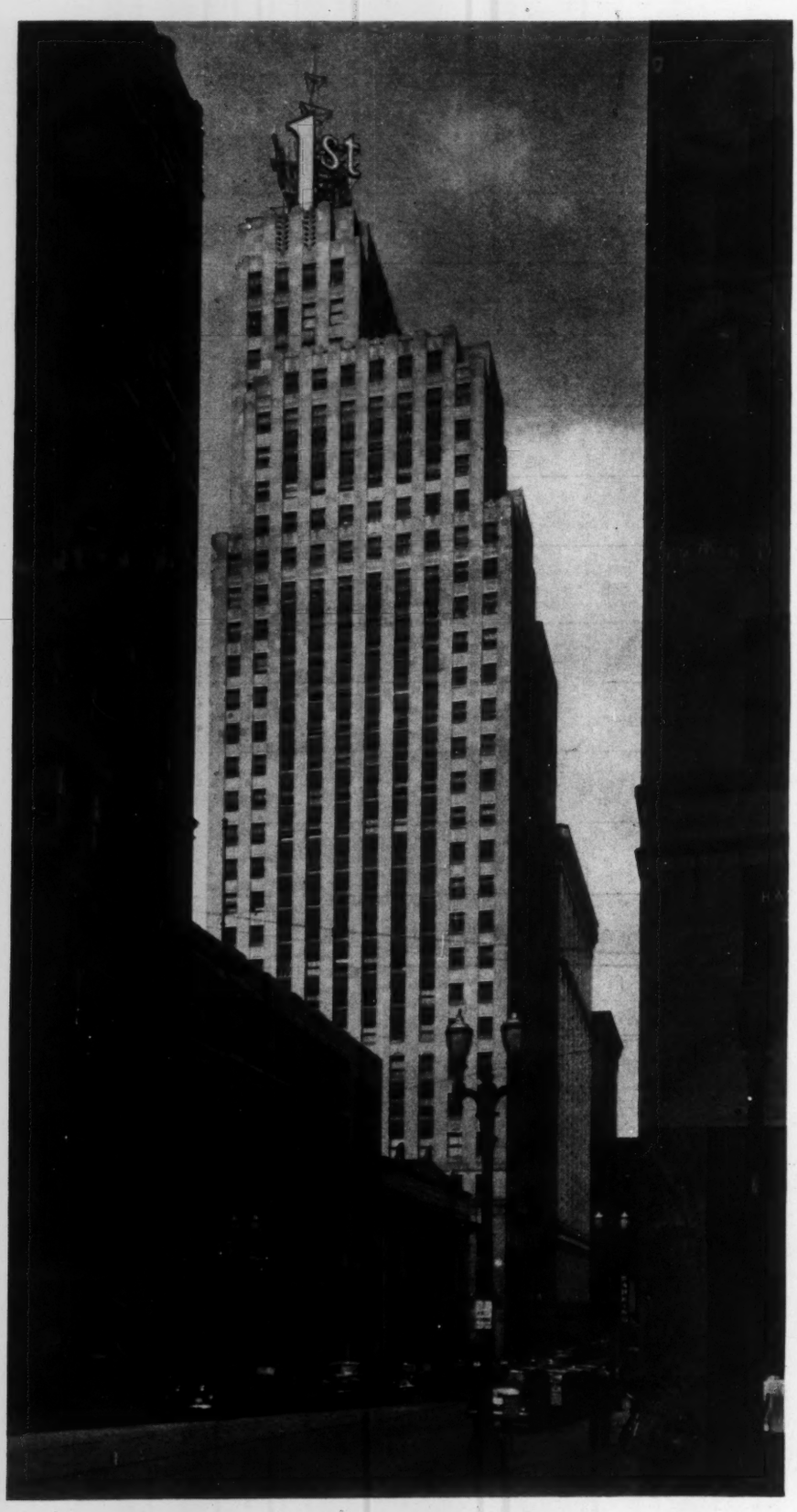
The cycle times listed in Table 1 were computed by means of Formula 7, and the off times listed in Table 1 were found by subtracting the on times from the cycle times.

[Next month Mr. Goodman will discuss open and closed circulating systems for storage jobs, mixing stored water and controls.]

Table 2—Water storage (Gallons per ton of compressor capacity)

Storage Capacity Btu	Allowable temperature rise of water in storage tank																									Storage Capacity Btu
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27
500	60	30	20	15	12	10	8.6	7.5	6.7	6.0	5.5	5.0	4.6	4.3	4.0	3.8	3.5	3.3	3.2	3.0	2.9	2.7	2.6	2.5	2.4	500
1000	120	60	40	30	24	20	17.2	15.0	13.3	12.0	10.9	10.0	9.2	8.6	8.0	7.5	7.1	6.7	6.3	6.0	5.7	5.5	5.2	5.0	4.8	1000
2000	240	120	80	60	48	40	34.3	30.0	26.7	24.0	21.8	20.0	18.5	17.2	16.0	15.0	14.1	13.3	12.6	12.0	11.4	10.9	10.4	10.0	9.6	2000
3000	360	180	120	90	72	60	51.4	45.0	40.0	36.0	32.7	30.0	27.7	25.7	24.0	22.5	21.2	20.0	19.0	18.0	17.2	16.4	15.7	15.0	14.4	3000
4000	480	240	160	120	96	80	68.6	60.0	53.4	48.0	43.7	40.0	36.9	34.3	32.0	30.0	28.2	26.7	25.3	24.0	22.9	21.8	20.9	20.0	19.2	4000
5000	600	300	200	150	120	100	85.8	75.0	66.7	60.0	54.6	50.0	46.2	42.9	40.0	37.5	35.3	33.3	31.6	30.0	28.6	27.3	26.1	25.0	24.0	5000
6000	720	360	240	180	144	120	103.0	90.0	80.0	72.0	65.5	60.0	55.4	51.4	48.0	45.0	42.4	40.0	37.9	36.0	34.3	32.7	31.3	30.0	28.8	6000
7000	840	420	280	210	168	140	120.0	105.0	93.4	84.0	76.4	70.0	64.6	60.0	56.0	52.5	49.4	46.7	44.2	42.0	40.0	38.2	36.5	35.0	33.6	7000
8000	960	480	320	240	192	160	137.0	120.0	106.0	96.0	87.3	80.0	73.9	68.6	64.0	60.0	56.5	53.4	50.5	48.0	45.7	43.7	41.8	40.0	38.4	8000
9000	1080	540	360	270	216	180	154.0	135.0	120.0	108.0	98.2	90.0	83.1	77.2	72.0	67.5	63.6	60.0	56.9	54.0	51.4	49.1	47.0	45.0	43.2	9000
10,000	1200	600	400	300	240	200	172	150	133	120	109	100	92.3	85.7	80.0	75.0	70.6	66.7	63.2	60.0	57.2	54.6	52.2	50.0	48.0	10,000
11,000	1321	660	440	330	264	220	189	165	147	132	120	110	102.0	94.3	88.0	82.5	77.7	73.4	69.5	66.0	62.9	60.0	57.4	55.0	52.8	11,000
12,000	1441	720	480	360	288	240	206	180	160	144	131	120	111.0	103.0	96.0	90.0	84.7	80.0	75.8	72.0	68.6	65.5	62.6	60.0	57.6	12,000
13,000	1561	780	520	390	312	260	223	195	173	156	142	130	120.0	111.0	104.0	97.5	91.8	86.7	82.1	78.0	74.3	70.9	67.9	65.0	62.4	13,000
14,000	1681	840	560	420	336	280	240	210	187	168	153	140	129.0	120.0	112.0	105.0	98.9	93.4	88.5	84.0	80.0	76.4	73.1	70.0	67.2	14,000
15,000	1801	900	600	450	360	300	257	225	200	180	164	150	139	129	120	113	106	100	94.8	90.0	85.7	81.8	78.3	75.0	72.0	15,000
16,000	1921	960	640	480	384	320	274	240	213	192	175	160	148	137	128	120	113	107	101.0	96.0	91.5	87.3	83.5	80.0	76.8	16,000
17,000	2041	1020	680	510	408	340	292	255	227	204	186	170	157	146	136	128	120	113	107.0	102.0	97.2	92.8	88.7	85.0	81.6	17,000
18,000	2161	1080	720	540	432	360	309	270	240	216	196	180	166	154	144	135	127	120	114.0	108.0	103.0	98.2	94.0	90.0	86.4	18,000
19,000	2281	1140	760	570	456	380	326	285	253	228	207	190	175	163	152	143	134	127	120.0	114.0	109.0	106.0	99.2	95.0	91.2	19,000
20,000	2401	1200	800	600	480	400	343	300	267	240	218	200	185	171	160	150	141	133	126	120	114	109	104	100	96.0	20,000
21,000	2521	1261	840	630	504	420	360	315	280	252	229	210	194	180	168	158	148	140	133	126	120	115	110	105	101.0	21,000
22,000	2641	1321	880	660	528	440	377	330	293	264	240	220	203	189	176	165	155	147	139	132	126	120	115	110	106.0	22,000
23,000	2761	1381	920	690	552	460	394	345	307	276	251	230	212	197	184	173	162	153	145	138	131	126	120	115	110.0	23,000
24,000	2881	1441	960	720	576	480	412	360	320	288	262	240	221	206	192	180	169	160	152	144	137	131	125	120	115.0	24,000
25,000	3001	1501	1000	750	600	500	429	375	333	300	273	250	231	214	200	188	177	167	158	150	143	136	130	125	120	25,000
26,000	3121	1561	1040	780	624	520	446	390	347	312	284	260	240	223	208	195	184	173	164	156	149	142	136	130	125	26,000
27,000	3241	1621	1080	810	648	540	463	405	360	324	295	270	249	232	216	203	191	180	171	162	154	147	141	135	130	27,000
28,000	3361	1681	1120	840	672	560	480	420	373	336	306	280	259	240	224	210	198	187	177	168	160	153	146	140	134	28,000
29,000	3481	1741	1160	870	696	580	497	435	387	348	316	290	268	249	232	218	205	193	183	174	166	158	151	145	139	29,000
30,000	3601	1801	1200	900	720	600	514	450	400	360	327	300	277	257	240	225	212	200	190	180	171	164	157	150	144	30,000
31,000	3721	1861	1240	930	744	620	532	465	414	372	338	310	286	266	248	233	219	207	196	186	177	169	162	155	149	31,000
32,000	3842	1921	1281	960	768	640	549	480	427	384	349	320	296	274	256	240	226	213	202	192	183	175	167	160	154	32,000
33,000	3962	1981	1321	990	792	660	566	495	440	396	360	330	305	283	264	248	233	220	209	198	189	180	172	165	158	33,000
34,000	4082	2041	1361	1020	816	680	583	510	454	408	371	340	314	292	272	255	240	227	215	204	194	186	177	170	163	34,000
35,000	4202	2101	1401	1050	840	700	600	525	467	420	382	350	323	300	280	263	247	233	221	210	200	191	183	175	168	35,000
36,000	4322	2161	1441	1080	864	720	617	540	480	432	393	360	332	309	288	270	254	240	227	216	206	196	188	180	173	36,000
37,000	4442	2221	1481	1110	888	740	635	555	494	444	404	370	342	317	296	278	261	247	234	222	212	202	193	185	178	37,000
38,000	4562	2281	1521	1140	912	760	652	570	507	456	415	380	351	326	304	285	268	253	240	228	217	207	198	190	182	38,000
39,000	4682	2341	1561	1170	936	780	669	585	520	468	426	390	360	334	312	293	275	260	246	234	223	213	204	195	187	39,000
40,000	4802	2401	1601	1200	960	800	686	600	534	480	437	400	369	343	320	300	282	267	253	240	229	218	209	200	192	40,000
41,000	4922	2461	1641	1230	984	820	703	615	547	492	447	410	379	352	328	308	290	273	259	246	234	224	214	205	197	41,000
42,000	5042	2521	1681	1261	1008	840	720	630	560	504	458	420	388	360	336	315	297	280	265	252	240	229	219	210	202	42,000
43,000	5162	2581	1721	1291	1032	860	737	645	574	516	469	430	397	369	344	323	304	287	272	258	246	235	224	215	206	43,000
44,000	5282	2641	1761	1321	1056	880	755	660	587	528	480	440	406	377	352	330	311	293	278	264	252	240	230	220	211	44,000
45,000	5402	2701	1801	1351	1080	900	772	675	600	540	491	450	416	386	360	338	318	300	284	270	257	246	235	225	216	45,000
46,000	5522	2761	1841	1381	1104	920	789	690	614	552	502	460	425	394	368	345	325	307	291	276	263	251	240	230	221	46,000
47,000	5642	2821	1881	1411	1128	940	806	705	627	564	513	470	434	403	376	353	332	313	297	282	269	256	245	235	226	47,000
48,000	5762	2881	1921	1441	1152	960	823	720	640	576	524	480	443	412	384	360	339	320	303	288	274	262	251	240	230	48,000
49,000	5882	2941	1961	1471	1176	980	840	735	654	588	535	490	452	420	392	368	346	327	310	294	280	267	256	245	235	49,000
50,000	6002	3001	2001	1501	1200	1000	858	750	667	600	546	500	462	429	400	375	353	333	316	300	286	273	261	250	240	50,000
51,000	6122	3061	2041	1531	1225	1020	875	765	680	612	557	510	471	437	408	383	360	340	322	306	292	278	266	255	245	51,000
52,000	6242	3121	2081	1561	1249	1040	892	780	693	624	567	520	480	446	416	390	367	347	329	312	297	284	271	260	250	52,000
53,000	6363	3181	2121	1591	1273	1060	909	795	707	636	578	530	489	454	424	398	374	353	335	318	303	289	277	265	255	53,000
54,000	6483	3241	2161	1621	1297	1080	926	810	720	648	589	540	499	463	432	405	381	360	341	324	309	295	282	270	259	54,000
55,000	6603	3301	2201	1651	1321	1100	943	825	733	660	600	550	508	472	440	413	388	367	348	330	314	300	287	275	264	55,000
56,000	6723	3361	2241	1681	1345	1120	960	840	747	672	611	560	517	480	448	420	395	373	354	336	320	306	292	280	269	56,000
57,000	6843	3421	2281	1711	1369	1140	978	855	760	684	622	570	526	489	456	428	403	380	360	342	326	311	298	285	274	57,000
58,000	6963	3481	2321	1741	1393	1160	995	870	774	696	633	580	536	497	464	435	410	387	366	348	332	316	303	290	279	58,000
59,000	7083	3541	2361	1771	1417	1180	1012	885	787	708	644	590	545	506	472	443	417	393	373	354	337	322	308	295	283	59,000
60,000	7203	3601	2401	1801	1441	1200	1029	900	800	720	655	600	554	514	480	450	424	400	379	360	343	327	313	300	288	60,000
61,000	7323	3661	2441	1831	1465	1220	1046	915	814	732	666	610	563	523	488	458	431	407	385	366	349	333	318	305	293	61,000
62,000	7443	3721	2481	1861	1489	1240	1063	930	827	744	677	620	573	532	496	465	438	413	392	372	354	338	324	310	298	62,000
63,000	7563	3782	2521	1891	1513	1261	1080	945	840	756	688	630	582	540	504	473	445	420	398	378	360	344	329	315	303	63,000
64,000	7683	3842	2561	1921	1537	1281	1098	960	854	768	698	640	591	549	512	480	452	427	404	384	366	349	334	320	307	64,000
65,000	7803	3902	2601	1951	1561	1301	1115	975	867	780	709	650	600	557	520	488	459	434	411	390	372	355	339	325	312	65,000
66,000	7923	3962	2641	1981	1585	1321	1132	990	880	792	720	660	609	566	528	495	466	440	417	396	377	360	344	330	317	66,000
67,000	8043	4022	2681	2011	1609	1341	1149	1005	894	804	731	670	619	575	536	503	473	447	423	402	383	366	350	335	322	67,000
68,000	8163	4082	2721	2041	1633	1361	1166	1020	907	816	742	680	628	583	544	510	480	454	430	408	389	371	355	340	327	68,000
69,000	8283	4142	2761	2071	1657	1381	1183	1035	920	828	753	690	637	592	552	518	487	460	436	414	394	377	360	345	331	69,000
70,000	8403	4202	2801	2101	1681	1401	1200	1050	934	840	764	700	646	600	560	525	494	467	442	420	400	382	365	350	336	70,000
71,000	8523	4262	2841	2131	1705	1421	1218	1065	947	852	775	710	656	609	568	533	501	474	449	426	406	387	371	355	341	71,000
72,000	8643	4322	2881	2161	1729	1441	1235	1080	960	864	786	720	665	617	576	540	508	480	455	432	412	393	376	360	346	72,000

MODERNIZES WITH AIR CONDITIONING



OF THE 430,000 sq ft of rentable and usable area in the First National Bank building, St. Paul, Minn., approximately 165,000 sq ft is air conditioned. All of this space was originally equipped with supply air fans with double bank spray washers and recirculating water pumps. This equipment gave fairly satisfactory results in fall, winter and spring, but in summer, temperature and humidity became excessive in most areas and particularly those with mass occupancy and high lighting intensities.

Upon completion of our new building, in 1931, I made a study of the possibilities of converting our washed air equipment to summer cooling. Fortunately, there are three artesian basins under our building, each carrying water at about 50 F. It appeared that if this water could be depended upon to produce a steady supply, it would be the cheapest cooling medium obtainable.

To satisfy ourselves of the permanency of this cold water supply, I made a study of the source of supply of each of these artesian basins and the amount of water being withdrawn from each in the Twin City area. The result of this study convinced us that we could count on the required supply for many years.

Suspended Matter in Well Water

We made a deal with a contractor to put down a 16 in. well guaranteed to produce 1000 gpm with not in excess of a 20 ft draw down and to produce water with not in excess of 10 parts per million of suspended matter. When the well was completed, after pumping 24 hr per day for seven days, tests showed a lowest suspended matter count of 50 parts per million. The contractor was about to give up and take a greatly reduced price for his job on account of excessive silica in the water, when it occurred to me that if the erosion caused by the rush of water into the hole and the whirling action of the water caused by

the action of the centrifugal pump could be neutralized, the inflow of silica could be stopped.

I suggested this to the contractors, who devised a clever method of counteracting these difficulties. They built a 6 in. pipe equal in length to the thickness of the sandstone strata (60 ft), slotted this pipe with ⅛ in. vertical slots all around and within 5 ft of each end, welded wings to the outside of the pipe to keep it in the center of the hole, set the pipe and packed it all around with pea gravel.

After this was completed they again started pumping at 1000 gpm and a test taken after 5 hr of pumping showed only seven parts per million, and after 10 hr, only three parts per million. Subsequent tests showed suspended matter as low as one part per million. By our foresight and persistence we secured water practically free from suspended matter, which was extremely important to future operation of our cooling equipment.

Changing System to Cooling

After the completion of the well and installation of the pumps, we ran new feed lines to all fan room areas of sufficient size to take care of the entire load and proceeded gradually to remodel the fan units for summer cooling. All of the old units were two bank spray washers, with recirculating pumps, and without filters. Typical procedure was about as follows:

One bank of sprays and moisture baffles were removed, and blast coils installed in about the position of the moisture baffles. Ahead of the one bank of remaining sprays we installed filter frames. Draft gage nipples were installed ahead and behind the filters to permit checking static pressure on filters; when the pressure reaches ½ in. the filters are changed. Each feed and return line is equipped with a thermometer to check efficiency of the coils. Water flow is controlled manually, and air temperatures are regulated by dampers.

MODERNIZES—Mr. Tyler, rental manager of the First National Bank building, St. Paul, Minn., described the modernization of the building with summer cooling at the recent Chicago convention of the National Association of Building Owners and Managers. The work involved converting the existing washed air ventilating system to summer cooling, and as underground water was available at 50 F, a well was put down. Difficulty was encountered due to the high amount of suspended matter in the water but was corrected in a clever way, as described by the author. He also tells how the fan units were changed over for cooling. . . . The building has recently installed auxiliary dampers in all outside supply and exhaust air openings, and finds that by doing so an annual saving of $1494 in steam for heating is made

Existing distribution ducts and grilles were not designed for cooling. At first, we had considerable trouble with cold drafts, but with minor adjustments in duct lines and installation of diffusing grilles we soon had almost everybody happy.

Since the system was originally designed for ventilation only, it does not deliver sufficient volumes of air to cool the southwest offices on the second and third floors, which have large windows and an extremely heavy sun load. We met this by installing recirculating cooling units.

The total cost of the conversion job ran about $30,000, including well, pumps and feed lines to the 12th floor for future development. The well water coils have a total capacity of about 300 tons. Operation costs run about $1000 a year, which represents electric current and repairs. No extra maintenance labor was involved.

Dampers Save $1494 a Year

An economy idea that we have just completed might not be amiss to mention—the installation of auxiliary dampers in all outside supply and exhaust air openings. These openings are on the fifth and ninth floor levels, a total of about 150 sq ft. With a 30 story building, and openings at these levels, the suction on them in cold weather is terrific. They were originally closed with regulation metal dampers.

Studying our daily steam report for one Sunday, when fans were not operating, I noticed that steam consumption per degree day was about the same as on a business day with the fans running. Investigation of this phenomenon showed that most of these metal dampers would not close tightly and that the inrush of air through the dampers on a cold day was sufficient in some cases to run the fans backward and to require steam in the coils at all times. Even though we could succeed in closing these metal dampers tight we would be in worse trouble, because in cold weather condensation would freeze them tight.

The solution arrived at after a lot of experimentation was the installation of a second set of dampers behind the metal dampers. The new dampers are built of wood and set in wood frames, with rubber gaskets in the rabbets. Each opening is divided into convenient operating size. The dampers are built of outside layers of ⅜ in. plywood with 1¼ in. insulation in between and a pine frame around the perimeter. The doors and frames are coated with asphalt paint and one coat of aluminum paint. Operation is with air motors of the same type as used on metal dampers, controlled so as to open and close with the starting and stopping of the fan motor.

This work was completed in December at a cost of about $2000. Steam consumption for January through May was 240 lb per degree day less than before secondary dampers were installed. This represents an annual saving of 1,800,000 lb of steam, or at our average cost of purchased steam, $1494.

Building Converts Washed Air Ventilating System to Summer Cooling, Using Well Water . . . By E. A. Tyler

HOTEL ENGINEERING

How to Save Electricity

ELECTRIC current cost in hotels is three times greater than the cost of water and like the water expense, the cost of electric current is increasing. This increase is not due to higher electric rates, but is the result of increased electric services—such as extensive air conditioning, more elaborate illumination facilities and electric cooking. These increases in the costs and consumptions of electric current are necessary if hotels wish to maintain their high standards of service and keep up with the times. However, many hotel operators have been able to offset these increased costs through the introduction of more efficient electrical equipment combined with the elimination of waste.

A study was issued last month by the American Hotel Association of the United States and Canada, 221 W. 57th St., New York, N. Y., which enables hotels to make practical comparisons of their electric costs and consumptions with those of other hotels similar in size and type and having approximately the same electric consuming facilities. This study is volume No. 2 of a series on hotel engineering problems prepared under the supervision of Georges C. St. Laurent, of Wegg & St. Laurent, hotel consultants. Comprising forty-four 8½x11 in. pages, it is available from the AHA at $1.50 per copy. In order that hotels might make fair and accurate comparisons of their electric results with those of other hotels, the study has arranged the figures on 324 hotels in six different tables, with each table containing the electric results of those hotels having similar electric consuming facilities.

The book describes how a hotel operator may conduct his own survey of the electric operation in his hotel and lists on a number of pages, various ideas for eliminating waste and lowering electric expense. The proper approach to the electric company rate problem is discussed, together with other important problems, such as electric generation compared with purchased electric current, and electric cooking compared with gas cooking.

Air Conditioning

In discussing guest room air conditioning units, it is stated that a 1 ton unit having a 1 hp motor requires about 1500 to 2000 watts in starting or about 14 to 18 amp of electric current from a 110 volt circuit. This is far in excess of what the average guest room electric circuit is capable of carrying in addition to that required for lighting. Therefore, where console units of more than ½ ton capacity are installed, it is usually necessary to provide special wiring for the ¾ or the 1 hp motors required to operate the ¾ or the 1 ton units. In order to avoid the expense of this special wiring, which may add as much as 20 per cent to the cost of the unit installed, it is advisable to make a careful check of the actual air conditioning load requirements of each guest room to be air conditioned.

A section of the study is devoted to ways of saving electric current, including that for air conditioning equipment operation. Among the recommendations given are that all doors and windows of air conditioned spaces be kept closed and windows facing the outside should be equipped with awnings or venetian blinds. Hotel operators should check over their air conditioning installations to find out how much outside air is being introduced, particularly as load requirements may have changed due to the room being used for different purposes, and provisions for outside air make-up provided in the original installation may exceed present requirements.

Such items as clean spray nozzles and air filters, an adequate supply of condenser cooling water with the proper temperature difference in entering and leaving the equipment, a sufficient supply of refrigerant and correct adjustment of the automatic control devices, are a few of the items that require careful and regular attention in the efficient operation of an air conditioning system, according to the report.

Ventilation Equipment

A substantial amount of electric current is consumed by practically every hotel for ventilation purposes, says the study. Furthermore, most hotels are built with elaborate ventilation systems designed to take care of the heaviest ventilating requirements. These systems are of two types: (1) those that are operated daily during certain hours or continuously, such as for the kitchen, dining room, basement and other spaces; and (2) those systems that are used only on certain occasions when banquet and special function rooms are to be occupied.

Because of the extensive amount of ventilation equipment in hotels, its operation should constitute a major responsibility for the hotel's engineering department as well as of the various departments—such as food—utilizing such facilities.

It is not unusual for ventilation fans to have motor drives of 15 or 20 hp, consuming 11,250 to 15,000 watts of electric current each, it is stated. Consequently, when one of these fans is left running unnecessarily, it is the same as leaving six hundred 25 watt lights burning without reason.

The operating hours for every fan should be scheduled with a log showing when and by whom each fan is started and stopped. A careful study should be made to see whether the operation of a fan is required or not.

Particular care should be exercised during the winter months not to operate fans except when required, because supply fans bring cold air into the building that must be preheated and exhaust fans discharge costly warm air to the outside.

Heating-Piping-Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

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Panel Heating and Cooling Analysis

By B. F. Raber* and F. W. Hutchinson**
Berkeley, Calif.

SUMMARY—A rational method of analyzing panel heating problems is described. General equations are given leading toward a determination of the mean radiant and inside temperatures corresponding to a fixed panel temperature and a known outside temperature. The intent of the paper is to outline the basic theory of radiant heating or cooling calculation rather than develop practical design methods.

RADIANT heating or cooling for comfort differs from convective heating or cooling primarily in that it attempts to control body heat loss through adjustment of the inside surface temperature rather than by adjustment of the room air temperature. Since only a fraction of the interior surface can be heated or cooled, the temperature of the controlled surface is a function of, but is not equal to, the weighted average or mean radiant temperature (MRT) of the room. Likewise the room air temperature reaches an equilibrium value which is in part determined by the panel temperature.

For the reasons given, the calculations involved in the exact design of a new radiant system or in estimating the characteristics of an existing radiant system are more complex than those needed in convective design. For convective systems an inside design air temperature is selected and a straightforward calculation determines the necessary rate of heat input or removal as a function of the outside temperature; the calculation is simple and direct, also exact within the limits of accuracy of the physical constants.

In panel heating, such a simple, direct and comparably exact solution does not seem feasible. The conditions necessary for comfort dictate a particular relationship between room air temperature (t_a) and MRT, yet neither t_a nor MRT is subject to direct control. For a particular system with a given ventilation rate and panel area, the panel surface temperature is the only variable subject to such control and it must vary as a function of the outside temperature. The basic design problem for such a system is, then, to select a panel surface temperature (t_p) such that the two terms t_a and MRT, (which vary with t_p), will reach such equilibrium values as will retain, with respect to one another, a relationship that will satisfy the criteria for comfort.

The function $t_a = \phi$ (MRT) which correlates conditions of substantially equal comfort has been established experimentally over a limited operating range. It is based essentially on physiological considerations, but as better data become available on the way in which the human body utilizes the three mechanisms of heat dissipation, evaporation, convection and radiation, a rational approach based on a heat balance at the skin may become possible. Lacking a more adequate explanation, $t_a = \phi$ (MRT) may be regarded as an empirical special condition arbitrarily imposed on conditions in the room.

But t_a and MRT must have specific values such that a heat balance will be established on the ventilation air and at any part of the interior surface. These equilibrium temperatures can be directly calculated from the heat transfer equations and are obviously dependent on the character of the surfaces and the conductances of the walls, floors, and other units of the construction.

Calculation Procedure

Thus for a given room with specified ventilation rate and panel area and a given outside temperature, the necessary panel surface temperature for comfort can be determined, by trial and error, as follows:

1. Assume a panel surface temperature and calculate the corresponding values of t_a and MRT.
2. From the comfort equation $t_a = \phi$ (MRT), using the value of t_a calculated in step 1, determine the value of MRT necessary for comfort.
3. If the values of MRT as calculated in steps 1 and 2 are the same, comfort will be realized at the assumed panel temperature. If the calculated MRT is below that for comfort the room will be too cold; if above, too hot; in this event assume a second value of t_p and repeat the analysis.

From the outline given, a particularly important fact becomes evident: the air temperature corresponding to comfort conditions in a radiant conditioned room will have a different equilibrium value for each value of the outside temperature. Thus, exact control of comfort in such a room cannot be realized by maintaining a constant inside temperature. As a corollary, simplified design methods which propose calculation of the necessary MRT in terms of a specified room air temperature are inherently incorrect since t_a is a variable of the system which can be fixed only in terms of the same variables—outside and panel temperatures—which are involved in the determination of MRT.

Therefore, the criterion of panel design is the condition that the environment be such as to produce equilibrium values of t_a and MRT which will fall on the curve $t_a = \phi$ (MRT), but it is not possible to specify the point on the curve at which the necessary condition will be realized. Fig. 1 shows the two conditions which must be satisfied. As the panel surface temperature is varied the slope of the heat transfer lines varies, but for any given value of t_p there is one and only one value of t_a for which both the heat transfer and comfort relationships are satisfied.

In setting up the equations for the rational design of a panel system the most difficult problem is to determine the mean radiant temperature. If the entire interior surface of the room were a panel and could be controlled at a constant and uniform temperature, that temperature would be equal to the MRT. The average panel in an actual room occupies only a small percentage of the total interior surface. Unheated parts of the surface reach different equi-

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librium values depending on the rates of convective and conductive loss or gain, and the position of the surface with respect to the panel and to other unheated surfaces. Since the MRT represents a uniform temperature at which the average radiant flux would be equivalent to that in the actual room, it follows that determination of the MRT requires a knowledge of surface temperatures at all points in the room. The basic engineering problem therefore simplifies to one of determining the surface temperature at any given point in the room.

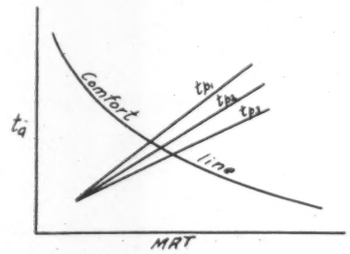

Fig. 1—Equilibrium values of air temperature and MRT at intersections with panel temperatures

The four sections of this paper deal with the classification and development of the equations and methods needed in the rational solution of panel design problems. The principal emphasis of this paper is on method of exact treatment. Such a theoretical analysis is justified for at least three eminently practical reasons:

1. Progress in the accurate design of radiant systems depends upon the development of more exact *short cut* methods which will more closely approximate the true condition of the actual system. While the exact analysis is in many instances too burdensome to be of direct application it may nonetheless effectively serve to indicate the direction in which development of approximate methods should proceed.

2. The theoretical analysis is of great practical utility in that it offers a basis of comparison with other methods and thereby directs attention to the effects of such assumptions as are customarily made in practice; it provides a reference point from which the accuracy of simplified solutions can be estimated. Every problem in the engineering research of radiant systems can be located, visualized and evaluated in the basic rational equations. In general, each panel research project has as its aim the simplification or evaluation of a single term or factor in the general theoretical equation.

3. The theoretical equations provide a relatively simple method of evaluating the boundary conditions when estimating the error likely to result from use of an inexact method.

The following development of methods for rational analysis of panel problems is divided into four sections starting with expressions for radiation from a single surface and proceeding to the general case of a heat balance on a radiant conditioned room with floor, ceiling, partitions, windows and exterior wall surfaces all subject to different heat balances. These sections are:

I. *Energy Leaving a Surface by Radiation:* This section brings together the radiation equations used in calculating the gross and net energy leaving a surface for various special cases and for the general case. The simpler equations are merely defined, but derivations are given for the less common reflected energy equations.

II. *Heat Balance on an Element of an Area:* In section II the equations of the preceding section are applied to the basic problem of determining the surface temperature at any point in various types of ideal systems. Numerical examples are included to show the error which can occur if the equations for a simple case are erroneously applied to an actual room.

III. *Average Surface Temperature as a Function of Average Shape Factor.* This section investigates the adequacy of the usual assumption that the average equilibrium temperature of a surface can be directly calculated from a knowledge of the average shape factor of that surface with respect to the surface from which energy is being received.

IV. *Determination of Equilibrium Conditions in a Radiant Conditioned Room:* The exchange equations of section II are here applied to the problem of calculating the average temperature of each different type of room surface and the inside air temperature. From the calculated surface temperatures the MRT can be evaluated and comfort conditions then checked from a curve similar to that of Fig. 1.

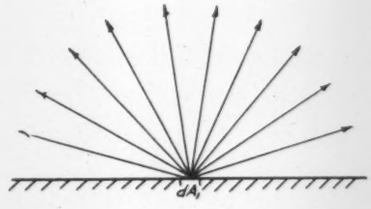

Fig. 2—For conditions under I (A)

I. *Energy Leaving a Surface, A_1, by Radiation*

In the cases which follow the temperature at a point on the surface A_1 is taken as constant with respect to time, but variable with respect to position. The emissivity, e_1, is taken as constant with respect to both time and position and is considered independent of both wave length and angle of emission. All surfaces considered in the analysis are perfect diffusers.

A) Total energy emitted,
By infinitesimal area dA_1

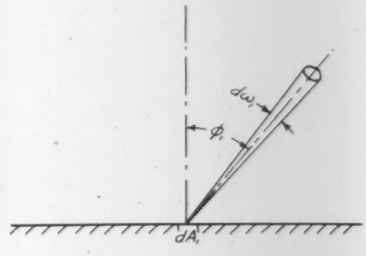

Fig. 3—For conditions under I (B)

$$dq = \sigma e_1 T_1^4 dA_1 \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

By area A_1

$$q = \sigma e_1 \int_{A_1} T_1^4 dA_1 \quad \ldots\ldots\ldots\ldots (2)$$

where q = Btu per hour.
A = square feet.
σ = Stefan - Boltzman constant = 0.172×10^{-8}.
T = degrees Fahrenheit absolute.

B) Energy emitted by the infinitesimal area dA_1 through the small solid angle $d\omega_1$ and in a direction making an angle ϕ_1 with the normal to dA_1.

$$d^2q = \sigma e_1 T_1^4 dA_1 \frac{\cos \phi_1 d\omega_1}{\pi} \quad \ldots\ldots\ldots\ldots (3)$$

since $\cos \phi_1 dA_1$ is the projected area of dA_1 normal to the direction of radiation and $\frac{d\omega_1}{\pi}$ is the fraction of hemispherical solid angle through which radiation occurs.

C) Energy leaving A_1 and striking A_2 when $e_2 = 1$.

1. When all other objects which A_1 *sees* are perfect absorbers ($e = 1$).

a) The energy emitted by dA_1 and striking dA_2,

$$d^2q = \sigma e_1 T_1^4 \frac{\cos\phi_1 \cos \phi_2 dA_1 dA_2}{\pi r^2} \ldots (4)$$

from Equation (3) noting that,

$$d\omega_1 = \frac{\cos \phi_1 \, dA_2}{r^2} \ldots\ldots\ldots\ldots (5)$$

b) The energy emitted by dA_1 and striking A_2,

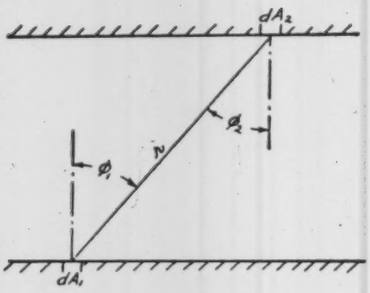

Fig. 4—For conditions under I (C)

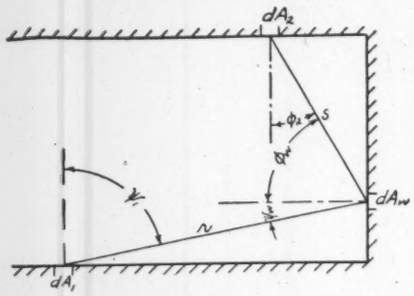

Fig. 5—For conditions under I (C) 2b

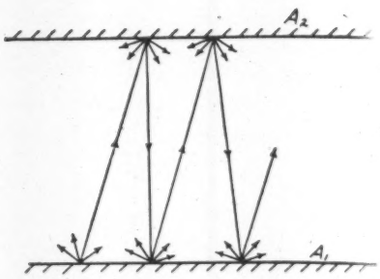

Fig. 6—For conditions under I (D)

$$dq = \sigma c_1 T_1^4 dA_1 \int_{A_2} \frac{\cos\phi_1 \cos\phi_2 \, dA_2}{\pi r^2} \quad \ldots\ldots (6)$$

where $\cos\phi_1$, $\cos\phi_2$ and r are variables with respect to the position of dA_1 on A_1. (For parallel surfaces—as in Fig. 4 — $\phi_1 = \phi_2$).

c) The energy emitted by A_1 and striking A_2 is then,

$$q = \sigma c_1 \int_{A_1} \int_{A_2} \frac{T_1^4 \cos\phi_1 \cos\phi_2 \, dA_1 dA_2}{\pi r^2} \quad \ldots\ldots (7)$$

2. When A_1 is completely enclosed by surfaces all of which, except A_2, are perfect reflectors ($c = o$).

a) Energy emitted by A_1 and striking A_2 as a result of direct passage from A_1 without reflection; given by Equations (4), (5), or (7).

b) Energy emitted by A_1 and striking A_2 after one reflection from a wall,

1. Energy emitted by A_1 and striking dA_w (from Equation (6),

$$dq = \sigma c_1 \, dA_w \int_{A_1} \frac{T_1^4 \cos\psi_1 \cos\psi_w \, dA_1}{\pi r^2} \quad \ldots\ldots (8)$$

2. Energy reflected by dA_w and striking dA_2,

$$d^2q = \sigma c_1 dA_w \frac{\cos\phi_w \cos\phi_2}{\pi s^2} \int_{A_1} \frac{T_1^4 \cos\psi_1 \cos\psi_w dA_1}{\pi r^2} dA_2 \quad \ldots\ldots (9)$$

3. Energy reflected by A_w and striking A_2 (= energy emitted by A_1 and striking A_2 after one reflection),

$$q = \sigma c_1 \int_{A_2} \int_{A_w} \left(\frac{\cos\phi_w \cos\phi_2}{\pi s^2} \right) \left(\int_{A_1} \frac{T_1^4 \cos\psi_1 \cos\psi_w dA_1}{\pi r^2} \right) dA_w \, dA_2 \quad \ldots\ldots (10)$$

c) Total energy striking A_2 from A_1, including an infinite number of wall reflections: Can be expressed as an integral equation or developed by a continuation of the method used in part "b". As will be shown in a later section, the evaluation of terms beyond the first reflection is in many practical cases unnecessary.

D) The total energy leaving A_1 and striking A_2 (including reflections from A_2 of energy initially emitted by A_1, but not including energy reflected by A_1 which is emitted by any other source).

1. When all other objects which A_1 sees have $c = 1$.

a) When $c_1 = 1$: Use Equation (7).

b) When $0 < c_1 < 1$,

$$q = \frac{\sigma c_1}{\pi} \int_{A_2} (a + kb + k^2 c + k^3 d + \ldots\ldots) \, dA_2 \quad \ldots\ldots (11)$$

(See derivation in Appendix A) *where,*

$$k = \frac{(1 - c_1)(1 - c_2)}{\pi^2}$$

$$a = \int_{A_1} \frac{T_1^4 \cos\phi_1 \cos\phi_2 \, dA_1}{r^2}$$

$$b = \int_{A_1} \left[\frac{\cos\phi_1 \cos\phi_2}{r^2} \right] \left[\int_{A_2} \left(\frac{\cos\phi_1 \cos\phi_2}{r^2} \right) (\text{Term "a"}) \, dA_2 \right] dA_1$$

2. When all other objects which A_1 sees are perfect reflectors ($c = o$).

a) Energy striking A_2 as a result of direct passage (including all direct reflections) from A_1 to A_2: Use Equation (11).

b) Energy striking A_2 after one reflection from a wall and without limit to the number of reflections between A_1 and A_2 directly. Combining Equations (10) and (11) after revising (11) to get the energy leaving dA_w

$$q = \frac{\sigma c_1}{\pi} \int_{A_2} \int_{A_w} \left[\frac{\cos\phi_w \cos\phi_2}{\pi s^2} \right] \left[\int_{A_1} \left(\frac{\cos\psi_1 \cos\psi_w}{\pi r^2} \right) (T_1^4 + ka' + k^2 b' + \ldots.) dA_1 \right] dA_w \, dA_2 \ldots (12)$$

$$a' = \int_{A_2} \left[\frac{\cos \phi_1 \cos \phi_2}{r^2} \right] \left[\int_{A_1} \frac{T_1^4 \cos \phi_1 \cos \phi_2 \, dA_1}{r^2} \right] dA_2$$

$$b' = \int_{A_2} \left[\frac{\cos \phi_1 \cos \phi_2}{r^2} \right] \left[\int_{A_1} \left(\frac{\cos \phi_1 \cos \phi_2}{r^2} \right) \left(\text{Term } a' \right) \, dA_1 \right] dA_2$$

c) Energy striking A_2 after multiple wall reflections: Energy will be returned to A_1 from A_2 by way of wall reflections; the quantity so returned must be added to the right side of Equation (11) to get the total energy from A_1 and this total can then be used in an expression of the same form as Equation (12) but carried out to n reflections. The equation for this case can be set up in the form of an integral equation, but its solution is so complex that the expression is of little utility.

II. *Heat Balance on an Element of Area, dA_1*

The element of area dA_1 is exchanging energy by radiation with a surface A_2 which is at a constant and uniform temperature T_2, by convection with room air at temperature T_a and by conduction and convection through the slab with outside air at temperature T_o: all temperatures are expressed in degrees Fahrenheit absolute. Conduction along the slab is taken as zero.

A) When $e_1 = e_2 = 1$ and dA_1 sees only A_2 and absolute space (equivalent to seeing a perfectly absorbing surface at a temperature of absolute zero.)

1. The net radiant gain of energy at dA_1 is equal to the energy absorbed at dA_1 from A_2 less the total energy emitted by dA_1, and from Equations (1) and (4),

$$= \sigma T_2^4 dA_1 \left(\int_{A_2} \frac{\cos \phi_1 \cos \phi_2 \, dA_2}{\pi r^2} \right) - \sigma T_1^4 \, dA_1 \quad \ldots\ldots\ldots\ldots\ldots\ldots (13)$$

The term under the integral is a function only of the geometry of the system and, for convenience, the following definition is used,

$$F_{A2,\,dA1} = \int_{A_2} \frac{\cos \phi_1 \cos \phi_2 \, dA_2}{\pi r^2} \quad \ldots (14)$$

where $F_{A2,dA1}$ is the *shape factor* and is equal to $\frac{A_2}{dA_1}$ multiplied by the fraction of energy leaving A_2 which falls on dA_1. Equation (13) then becomes,

$$= \sigma dA_1 \left(T_2^4 F_{A2,\,dA1} - T_1^4 \right) \quad \ldots\ldots (15)$$

where the shape factor can be evaluated for any particular system.[1]

2. Heat balance on dA_1:

$$\sigma dA_1 \left(T_2^4 F_{A2dA1} - T_1^4 \right) = h_1 dA_1 (T_1 - T_a) + C_1 dA_1 (T_1 - T_o) \quad \ldots\ldots (16)$$

where

h_1 = film coefficient of heat transfer from surface dA_1 to room air.

C_1 = conductance of slab and outside air film from surface dA_1 to outside air.

3. Numerical example. Consider a slab of area A_1 uniformly heated to a surface temperature of 110 F and located at distance L opposite and parallel to an unheated slab of equal area. The unheated surface is perfectly insulated, conduction term drops out of Equation (16), and transfers heat by convection with room air at 60 F. The film transfer coefficient is taken as 0.75 (the commonly used film coefficient of 1.65 includes radiant as well as convective heat transfer; it should not be used in radiant heating problems).

Equation (16) can be used to determine the equilibrium temperature at various points on the unheated surface. Figs. 8a, b, c, show equilibrium temperatures at the center, middle of each side and corner, of the unheated slab for different arrangements. Note that the temperatures shown are determined by the geometry of the arrangement rather than by the actual dimensions of the surface. Thus in Fig. 8a the temperatures would remain the same if the surfaces were

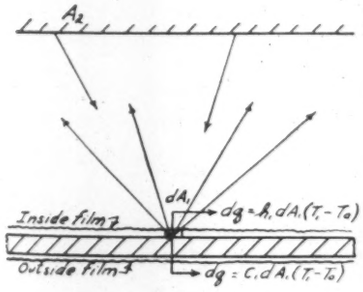

Fig. 7—For conditions under II (A) 2

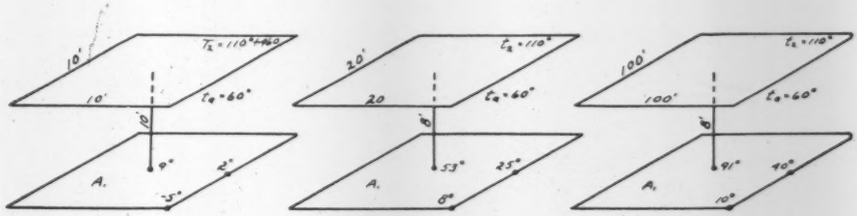

Fig. 8, a, b, c—For conditions under II (A) 3

[1] See *Heat Transmission* by McAdams.

20 ft by 20 ft or 100 ft by 100 ft, providing the distance between them is changed to 20 ft and to 100 ft, respectively.

In practice, a system in which A_1 would see absolute space could never be realized and equilibrium temperatures as low as those of Fig. 8 would not occur. However, if absolute space is replaced by a clear night sky, the nocturnal radiation from A_1 can be measured or can be roughly calculated by assuming an equivalent temperature of space as —50 F[2] and for such an actual system the equilibrium temperatures will remain lower than the temperature of the air in contact with the surface.

B) When $e_1 = e_2 = 1$ and dA_1 is enclosed by surfaces all of which, except A_2, are at the same temperature as A_1 and have $e = 1$. (These conditions define a system in which the only net radiant transfer to or from dA_1 is with respect to direct exchange with A_2.)

1. Heat balance. From Equations (6), (14), and (16) noting that $F_{A_2dA_1}dA_1 = F_{dA_1A_2}A_2$ obtain.... (17)

$$\sigma\ F_{A_2dA_1}dA_1\ (T_2^4 - T_1^4) = h_1 dA_1 (T_1 - T_a) + C_1 dA_1\ (T_1 - T_o) \ldots \textbf{(18)}$$

2. Numerical example. Using Equation (18) with the surface arrangements of Fig. 8, results in Fig. 9 a, b, c.

C) When $e_1 = e_2 = 1$ and dA_1 is enclosed by surfaces all of which, except A_2, are perfect reflectors. (These conditions define a system in which all energy emitted by dA_1 is absorbed by A_2 or is later reabsorbed by A_1 after reflection.)

Net transfer from A_2 to dA_1
= direct transfer without reflection (a in Fig. 10)
+ transfer as result of one reflection (b in Fig. 10)
+ transfer as result of two reflections (c in Fig. 10)
+ transfer as result of three reflections
+ transfer as result of n reflections

1. Direct transfer a can be evaluated from Equation (6) [it is equal to the radiation term in Equation (18)].

2. Transfer as a result of one reflection, b, is from Equation (10),

$$dq = \sigma\ (T_2^4 - T_1^4)\left[\int_{A_w}\left(\frac{\cos\psi_1 \cos\psi_w}{\pi r^2}\right)\left(\int_{A_2}\frac{\cos\phi_2 \cos\phi_w\, dA_2}{\pi s^2}\right)dA_w\right]dA_1 \ldots (19)$$

$$= \sigma\ (T_2^4 - T_1^4)\ dA_1 \int_{A_w}(F_{A_2dA_w})\left(\frac{\cos\psi_1 \cos\psi_w}{\pi r^2}\right)dA_w \ldots . (20)$$

For a small but finite area of the wall, ΔA_w, the shape factor $F_{A_2dA_w}$ remains substantially constant and can be taken out from under the integral sign to give,

$$dq = \sigma\ (T_2^4 - T_1^4)\ dA_1 \sum_{\Delta A_w}^{A_w}(F_{A_2\Delta A_w})\left(\int_{A_w}\frac{\cos\psi_1 \cos\psi_w\, dA_w}{\pi r^2}\right) \ldots \ldots \ldots . (21)$$

or

$$dq = \sigma\ (T_2^4 - T_1^4)\ dA_1 \sum_{\Delta A_w}^{A_w}(F_{A_2\Delta A_w})(F_{\Delta A_w dA_1}) \ldots . (22)$$

Equation (22) can be more simply written,

$dq = \sigma\ (T_2^4 - T_1^4)\ dA_1 \cdot F_{A_2dA_1r_1}$ ..... (23)

where $F_{A_2dA_1r_1}$ is $\frac{A_2}{dA_1}$ multiplied by the fraction of energy emitted by A_2 which strikes dA_1 after one wall reflection and it is defined by the equation,

$$F_{A_2dA_1r_1} = \sum_{\Delta A_w}^{A_w}(F_{A_2\Delta A_w})\ (F_{\Delta A_w dA_1}) \ldots \ldots \ldots \ldots \ldots \ldots \ldots . (24)$$

3. The transfer of energy resulting from further reflections can be expressed as,

a) For two reflections,

$dq = \sigma\ (T_2^4 - T_1^4)\ dA_1 \cdot F_{A_2dA_1r_2}$ ..... (25)

b) For n reflections,

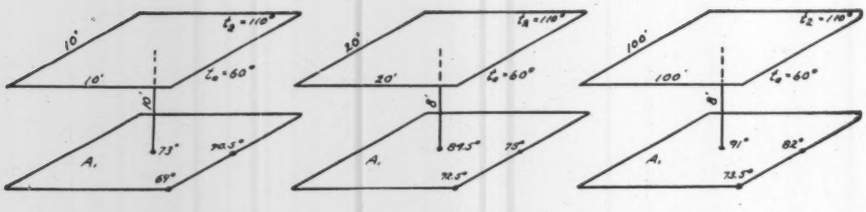

Fig. 9, a, b, c—For conditions under II (B) 2

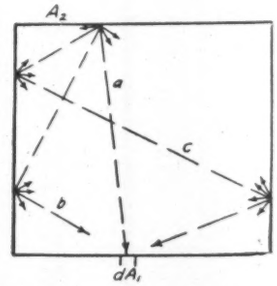

Fig. 10 — For conditions under II (C)

[2]See *The Calculation of Heat Transmission*, by Fishenden and Saunders.

$$dq = \sigma\ (T_2^4 - T_1^4)\ dA_1 \cdot F_{A2dA1rn} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (26)$$

where the terms $F_{A2dA1rn}$ can be determined by analysis as in *b* above.

4. The total energy transferred from A_2 to dA_1 is then,

$$dq = \sigma\ (T_2^4 - T_1^4)\ dA_1\ (F_{A2dA1} + F_{A2dA1r1} + F_{A2dA1r2} + \ldots\ldots\ldots + F_{A2dA1rn}) \quad \ldots . (27)$$

5. Numerical example. Fig. 11*a* shows the variation of shape factor for the direct transfer of energy from A_2 to A_1, first term in series of Equation (27); the system is a room 20 ft x 20 ft x 8 ft and is shown in elevation in Fig. 11*c*. Fig. 11*b* shows variation of the shape factor for energy which strikes A_1 after one reflection, second term in the series of Equation (27).

From the values plotted in Figs. 11*a* and 11*b* it is evident that the distribution of energy from A_2 and falling on dA_1 is extremely variable with respect to the position of dA_1 for energy which is transferred directly, but is practically uniform for energy received after one reflection. Since the degree of uniformity must increase as the number of reflections increases it follows that for this particular case a satisfactory and more direct solution can be obtained by considering that all energy except that which is directly transferred is uniformly received over A_1. With this assumption and taking the overall shape factor of A_1 with respect to A_2 as a constant having the calculable value K, Equation (27) becomes,

$$dq = \sigma\ (T_2^4 - T_1^4)\ dA_1 \left[F_{A2dA1} + \left(K - \frac{\int_{A_1} F_{A2dA1} dA_1}{A_1} \right) \right] \ldots (28)$$

If F_{A2dA1} is taken as constant over the small area ΔA_1,

$$dq = \sigma\ (T_2^4 - T_1^4)\ dA_1 \left[F_{A2dA1} + \left(K - \frac{\sum^{A_1} F_{A2\Delta A1}}{\frac{\Delta A_1}{A_1}} \right) \right] \ldots\ldots (29)$$

Evaluating terms of Equation (29) from the shape factors given in Fig. 11,

$$\frac{\sum^{A_1} F_{A2\Delta A1}}{\frac{\Delta A_1}{A_1}} = \frac{4\ (0.62 + 0.50 + 0.50 + 0.40)\ 25}{20 \times 20} = 0.51, \text{ so,}$$

$$dq = \sigma\ (T_2^4 - T_1^4)\ dA_1\ [F_{A2\Delta A1} + (K - 0.51)]$$

which can be written,

$$dq = \sigma\ (T_2^4 - T_1^4)\ dA_1 F_{A2\Delta A1e} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (30)$$

where $F_{A2\Delta A1e}$ is an equivalent shape factor, including reflections, for any particular element of area ΔA_1. Numerical values of $F_{A2\Delta A1e}$ at various points on A_1 are shown in Fig. 12*a* while the corresponding equilibrium temperatures, calculated from the heat balance equation,

$$\sigma\ (T_2^4 - T_1^4)\ dA_1 F_{A2\Delta A1e} = h_1 dA_1\ (T_1 - T_a) + C_1 dA_1\ (T_1 - T_o) \ldots . (31)$$

are shown in Fig. 12*b*. Of particular interest is the fact that the equivalent shape factor at the center of A_1 (Fig. 12*a*) would exceed unity if the walls were perfect specular reflectors (instead of reradiating matt surfaces from which the distribution of reflected energy is independent of angle).

D) When $e_1 = e_2 = 1$ and dA_1 is enclosed by surfaces all of which, except A_2, have e_w between zero and unity. The walls are considered to be at the same temperature as the particular element of area, dA_1, on which the heat balance is being established.

Using the method of analysis developed for perfectly reflecting walls, but noting that the actual wall reflectivity, $(1 - e_w)$, becomes a multiplying factor in Equation (9), it is found that Equation (27) becomes:

$$dq = \sigma (T_2^4 - T_1^4) dA_1\ [F_{A2dA1} + (1 - e_w) F_{A2dA1r1} + \ldots + (1 - e_w)^n F_{A2dA1rn}] \quad \ldots\ldots (32)$$

and the heat balance on dA_1 is therefore given by,

$$\sigma (T_2^4 - T_1^4) dA_1\ [F_{A2dA1} + (1 - e_w)\ F_{A2dA1r1} + \ldots + (1 - e_w)^n\ F_{A2dA1rn}] = h_1 dA_1 (T_1 - T_a) + C_1 dA_1 (T_1 - T_o) \ldots\ldots (33)$$

Equation (33) is not subject to treatment by the simplification of Equation (28) and its solution depends on evaluation of the series.

E) When e_1 and e_2 are each between zero and unity and A_1 is enclosed by surfaces all of which, except A_2, have $e_w = 1$ and are at a temperature equal to that of dA_1.

The net radiant heat transfer is evaluated in a manner similar to that

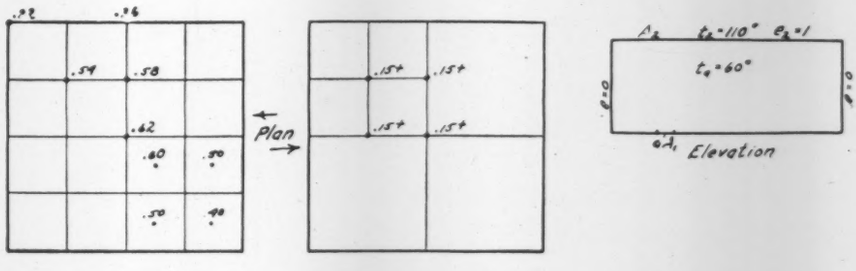

Fig. 11, a, b, c—For conditions under II (C) 5

used in deriving Equation (11); the resultant heat balance is:

$$\frac{\sigma c_1}{\pi}\left[T_2^4(a+kb+k^2c+\ldots)-T_1^4\int_{A_2}(a'+kb'+k^2c'+\ldots)dA_2\right]dA_1=h_1dA_1(T_1-T_a)+C_1dA_1(T_1-T_o)\ldots.(34)$$

$$where,\ a=\int_{A_2}\frac{\cos\phi_1\cos\phi_2 dA_2}{r^2}$$

$$a'=\frac{\cos\phi_1\cos\phi_2}{r^2}$$

$$b=\int_{A_2}\left[\frac{\cos\phi_1\cos\phi_2}{r^2}\right]\left[\int_{A_1}\left(\frac{\cos\phi_1\cos\phi_2}{r^2}\right)(\text{Term } a)dA_1\right]dA_2$$

$$b'=\int_{A_1}\left[\frac{\cos\phi_1\cos\phi_2}{r^2}\right]\left[\int_{A_2}\left(\frac{\cos\phi_1\cos\phi_2}{r^2}\right)(\text{Term } a')dA_2\right]dA_1$$

$$k=\frac{(1-c_1)\ (1-c_2)}{\pi^2}$$

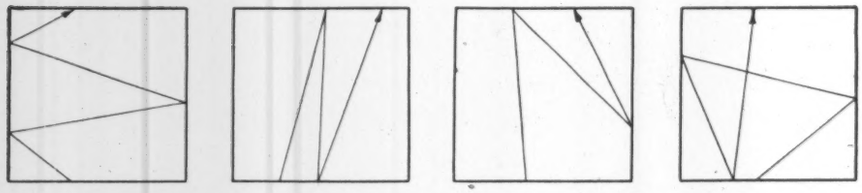

Fig. 13, a, b, c, d—For conditions under II (F)

F) When c_1, c_2 and c_w are all between zero and unity (not necessarily equal to one another) and A_1 is enclosed by surfaces all of which, except A_2, are at a temperature equal to that of the element of area dA_1 on which the heat balance is to be established.

Fig. 13 shows some of the paths followed by energy initially emitted by dA_1. Fig. 13*a* shows a typical path for that part of the energy emitted by dA_1 which reaches A_2 after one or more reflections from the wall surfaces; this fraction can be evaluated by a series of equations of the form of Equation (10). Fig. 13*b* shows path for the energy transferred as a result of reflections between A_1 and A_2; this fraction is evaluated in Equation (11). Figs. 13*c* and *d* show possible paths involving reflections between walls and either A_1 or A_2; evaluation of transfer by such paths is possible, but too complex to be of great practical value.

Since the system described more closely approaches the conditions of an actual room than do any of the preceding cases, it follows that an exact analytical solution of practical problems is usually not feasible; the practical problem must be solved by setting up equations for some less complex ideal system. The more closely the ideal system approaches the real system, the less will be the error. The section which follows will show by example the magnitude of the error which can result from over simplification of the actual system.

III. Average Surface Temperature as a Function of Average Shape Factor

A procedure commonly employed in practical problems is to calculate an average surface temperature based on an average value of the shape factor of that surface with respect to the surface with which it is transferring energy. The adequacy of this method can be investigated by examining any one of the heat balance equations. Consider Equation (18),

$$\sigma F_{A2dA1}\,dA_1(T_2^4-T_1^4)=h_1dA_1(T_1-T_a)+C_1dA_1(T_1-T_o)\quad\ldots(18)$$

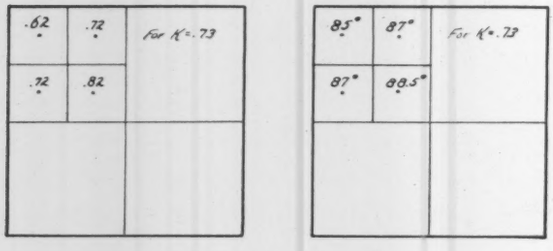

Fig. 12, a, b—For conditions under II (C) 5

If all terms in this equation other than T_1 and F_{A2dA1} are known, it becomes possible to calculate the equilibrium value of T_1 corresponding to any particular shape factor. From a series of such calculations data are obtained to plot T_1 as a function of the shape factor. The position and shape of the resultant curve (see Fig. 14) is dependent on the numerical value of the terms of

the heat balance equation, but for the present discussion the important fact is that T_1 is shown to be a non-linear function of the shape factor. This being so, the true average of point temperatures on the surface is not equal to the temperature corresponding to the average of point shape factors, or

$$\frac{\int_A F dA}{A} \neq \frac{K \int_A T dA}{A} \quad \ldots\ldots\ldots\ldots (35)$$

In the usual operating range of low temperature panels the curve of Fig. 14 so closely approximates the function, $T_1^4 = \phi(F_{A_2 dA_1})$ that the two curves may be considered as superimposed. Thus the mean effective radiation from A_1 is not a linear function of the shape factor or,

$$\frac{\int_A F dA}{A} \neq \frac{\int_A T^4 dA}{A} \quad \ldots\ldots\ldots\ldots (36)$$

so the temperature determined from the average shape factor is not the true average for the surface nor is it the equivalent temperature at which a uniformly heated surface would emit an amount of energy equal to the emission from the actual infinite number of point sources at different temperatures.

The determination of a curve similar to that of Fig. 14 is easily obtained for any specific set of conditions; its construction will eliminate need for the assumption of an equivalent average temperature, since the temperatures at points can be directly obtained from the curve (when shape factors are known) and the true equivalent temperature then calculated by the equation,

$$T_1 \text{ equivalent} = \frac{\sum_{\Delta A_1}^{A_1} T_1^4 \Delta A_1}{A_1} \quad \ldots\ldots (37)$$

Further, use of such a curve will show the magnitude of the error likely to result from the approximate method; in many cases the difference is not great enough to justify the added work of a rigorous solution.

IV. Determination of Equilibrium Conditions in a Radiant Conditioned Room

A) General statement of the problem.

In any room heated or cooled by radiant means each point on an unheated interior surface will reach an equilibrium temperature corresponding to particular and known values of the panel surface temperature, t_p and the outside air temperature, t_o; these equilibrium temperatures can be evaluated by setting up a heat balance at each surface and on the ventilation air as it passes through the room. Preceding sections have shown that the temperature on an unheated surface is not uniform, but varies with the extent to which each point on the surface sees the panel. Thus in a closed room there are an infinite number of unknown surface temperatures and an exact analysis of surface temperatures in such a room would require the simultaneous solution of an infinite number of equations.

While an exact solution is obviously impractical, a solution accurate to any prescribed limit can be obtained by breaking up the unheated interior surface into small areas ΔA and considering the surface temperature constant over each such area. The smaller ΔA becomes, the more exact will be the solution.

In many cases the most significant variation in surface temperature will occur when the conductance and film coefficient change and when the general position of the surface in question is drastically altered with respect to the panel. Where this is the case a substantially accurate analysis of the surface temperatures throughout the room can be made by taking ΔA as equal to the total area of each different type of surface; the analysis will then require simultaneous solution of as many heat balance equations as there are different surfaces.

The method of solution outlined will determine an average temperature of each inside surface. As has been shown in Section III, the average temperature calculated in this way is not exactly equal to the true average of the point temperatures across the surface, but the difference is sufficiently small to be neglected in many problems. Knowing the average temperatures of all surfaces (and the air temperature) it becomes possible to calculate point temperatures on any one of these surfaces. Continuing in this way isothermals can be located on all surfaces and values of the previously determined average temperatures corrected for the non-linear relationship between,

$$\frac{\int_A F dA}{A} \text{ and } \frac{\int_A T dA}{A}$$

Using the corrected surface temperature it is possible to calculate the mean radiant temperature. The mean radiant temperature and air temperature together establish the environment of the room; if the intersection of the calculated MRT and t_a temperature falls on the comfort line, when this line is plotted as MRT $= \phi(t_a)$ the panel will provide comfort. If the MRT and t_a lines do not intersect on the comfort line, a change in the heat transfer characteristics of the system will be necessary; this can be accomplished by changing the ventilation rate, panel area or panel surface temperature.

B) Determination of average surface temperatures.

Consider a room heated or cooled by one or more ceiling panels at constant and uniform surface temperature, t_p. The panels will see the following different types of surfaces: exterior wall, windows, floor, inside partitions; the unheated portion of the ceiling constitutes a sixth type of surface, but one that is not *visible* to the panels. No specification will

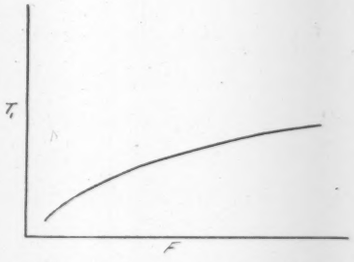

Fig. 14—For conditions under III

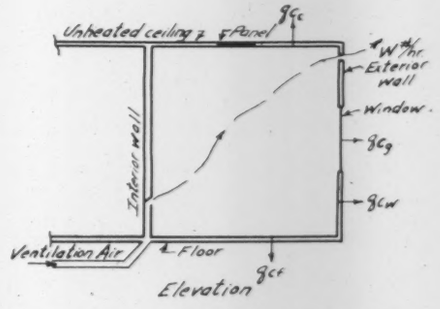

Fig. 15—For conditions under IV (B)

be made concerning the distribution of surfaces, but it will be considered that (except for panel and unheated ceiling) each surface is able to see five other types of surfaces. Surfaces of the same type will be considered as at the same temperature regardless of the fact that they may be in different walls. In the following treatment each type of surface will be treated as though it occurred only once in the system; for windows, etc., which appear in more than one wall the necessary extension of this treatment is obvious. The equations will be established for a room in which all surfaces have an emissivity of unity (for most surfaces found in practice the emissivity exceeds 0.9).

1. *Nomenclature.*

q = Btu per (hr) (sq ft)

First subscript:

C = conduction
V = convection
R = radiation
W = pounds per hour of ventilation air passing through the room.

Second subscript:

f = floor
w = outside wall
g = glass
c = unheated ceiling
i = interior wall adjoining heated room
p = panel
o = outside air
a = room air

Examples:

q_{cg} = Btu per (square foot) (hour) transfer by conduction between the inside surface of the glass and the outside air.

q_{Rpf} = Btu per (square foot) (hour) transfer by radiation from the panel to the floor, based on area of the floor.

h_f = film coefficient for heat transfer by convection from floor surface to air in room.

C_f = conductance from inside surface of floor to outside air.

2. Heat balance on each type of unheated surface.

a) Heat balance on the floor.

Net transfer to floor by radiation = transfer from floor by convection + transfer from floor by conduction.

or, from Fig. 16,

$$q_{Rif} + q_{Rpf} + q_{Rcf} + q_{Rgf} + q_{Rwf} = q_{vf} + q_{cf} \quad \ldots\ldots (38)$$

but by Equation (7),

$$q_{Rif} = \sigma(T_i^4 - T_f^4) \int_{A_i} \int_{A_f} \frac{\cos\phi_i \cos\phi_f \, dA_i dA_f}{\pi r^2} \quad \ldots\ldots (39)$$

$$= \sigma(T_i^4 - T_f^4) \, F_{if} A_f \quad \ldots\ldots (40)$$

$$F_{if} = \frac{1}{A_f} \int_{A_i} \int_{A_f} \frac{\cos\phi_i \cos\phi_f \, dA_i \, dA_f}{\pi r^2} \quad \ldots\ldots (41)$$

Since the convection and conduction terms of Equation (38) are expressed in terms of the first power of the temperature, it would simplify the solution if the radiation terms could also be written in terms of the first power. This can be accomplished by defining an equivalent transfer coefficient for radiation by the equation,

$$h_R = \frac{\sigma(T_i^4 - T_f^4)}{T_i - T_f} \quad \ldots\ldots (42)$$

$$q_{Rif} = \sigma(T_i^4 - T_f^4) \, F_{if} A_f = h_R (T_i - T_f) \, F_{if} A_f = h_R (t_i - t_f) \, F_{if} A_f \quad \ldots\ldots (43)$$

When approximate values of t_i and t_f are known or can be estimated, an accurate evaluation of h_R is possible—lacking such knowledge of temperatures h_R can be taken as approximately equal to 1.08; for the average temperature range of low temperature panel heating or cooling this value is accurate to within 4 per cent.

Substituting h_R in Equation (40), and taking h_R as substantially constant for the conditions of the room,

$$h_R A_i \Big[F_{if}(t_i - t_f) + F_{pf}(t_p - t_f) + F_{cf}(t_c - t_f) + F_{gf}(t_g - t_f) + F_{wf}(t_w - t_f) \Big]$$

$$= \Big[h_f (t_f - t_a) + C_f (t_f - t_o) \Big] A_f \quad \ldots\ldots (44)$$

$$t_f = \frac{h_R [F_{if} t_i + F_{pf} t_p + F_{cf} t_c + F_{gf} t_g + F_{wf} t_w] + h_f t_a + C_f t_o}{h_R (F_{if} + F_{pf} + F_{cf} + F_{gf} + F_{wf}) + h_f + C_f} \quad \ldots\ldots (45)$$

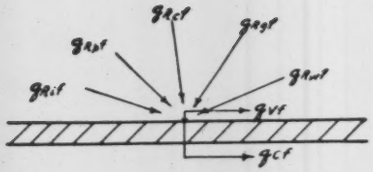

Fig. 16 — For conditions under IV (B) 2

Since the entire field of vision of the floor is taken up with the five surfaces whose shape factors with respect to the floor appear in the denominator of Equation (45), it follows that,

$$F_{if} + F_{pf} + F_{cf} + F_{gf} + F_{wf} = 1 \quad \ldots\ldots (46)$$

$$t_f = \frac{h_R (F_{if} t_i + F_{pf} t_p + F_{cf} t_c + F_{gf} t_g + F_{wf} t_w) + h_f t_a + C_f t_o}{h_R + h_f + C_f} \quad \ldots\ldots (47)$$

b) Heat balance on exterior wall surface.

By analogy with Equation (45),

$$t_w = \frac{h_R(F_{tw}t_t + F_{iw}t_i + F_{pw}t_p + F_{cw}t_c + F_{gw}t_g) + h_w t_a + C_w t_o}{h_R(F_{tw} + F_{iw} + F_{pw} + F_{cw} + F_{gw}) + h_w + C_w} \quad \ldots (48)$$

In this case, unlike that of the floor, the entire field of vision of the exterior wall may not be occupied by the other five surfaces, as there may be sections of exterior wall in more than one wall of the room. If some of this exterior wall is able to see another part of the same type of surface then,

$$F_{tw} + F_{iw} + F_{pw} + F_{cw} + F_{gw} = 1 - F_{ww} \quad \ldots (49)$$

where F_{ww} is the shape factor of exterior wall with respect to such portions of the same type of surface as may be in other walls of the room. Note that all shape factors for this case should be expressed in terms of the total area of the exterior wall surfaces. Then from Equation (48)

$$t_w = \frac{h_R(F_{tw}t_t + F_{iw}t_i + F_{pw}t_p + F_{cw}t_c + F_{gw}t_g) + h_w t_a + C_w t_o}{h_R(1 - F_{ww}) + h_w + C_w} \quad \ldots (50)$$

Note that if there is exterior wall surface in only one wall of the room the term $F_{gw}t_g$ drops out because the exterior wall is then unable to see the window.

c) Heat balance on glass. By analogy from Equation (50),

$$t_g = \frac{h_R(F_{wg}t_w + F_{tg}t_t + F_{ig}t_i + F_{pg}t_p + F_{cg}t_c) + h_g t_a + C_g t_o}{h_R(1 - F_{gg}) + h_w + C_w} \quad \ldots (51)$$

Note that if there is only one exterior wall in the room the term $F_{wg}t_w$ drops out because the window cannot then see the exterior wall.

d) Heat balance on unheated area of ceiling. By analogy from Equation (47) and noting that this balance includes one less term than the preceding ones because the unheated ceiling is unable to see the panel,

$$t_c = \frac{h_R(F_{gc}t_g + F_{wc}t_w + F_{tc}t_t + F_{ic}t_i) + h_c + t_a + C_c t_o}{h_R + h_c + C_c} \quad \ldots (52)$$

e) Heat balance on inside partition. By analogy from Equation (47),

$$t_i = \frac{h_R(F_{gi}t_g + F_{wi}t_w + F_{ti}t_t + F_{pi}t_p + F_{ci}t_c) + h_i t_a + C_i t_o}{h_R + h_i + C_i} \quad \ldots (53)$$

f) Heat balance on air passing through the room,

$$0.24W(t_a - t_o) = h_p(t_p - t_a)A_p + h_c(t_c - t_a)A_c + h_g(t_g - t_a)A_g + h_w(t_w - t_a)A_w + h_t(t_t - t_a)A_t + h_i(t_i - t_a)A_i \quad \ldots (54)$$

$$t_a = \frac{h_p t_p A_p + h_c t_c A_c + h_g t_g A_g + h_w t_w A_w + h_t t_t A_t + h_i t_i A_i + 0.24Wt_o}{h_p A_p + h_c A_c + h_g A_g + h_w A_w + h_t A_t + h_i A_i + 0.24W} \quad \ldots (55)$$

C) Determination of mean radiant temperature (MRT).

From section B six independent Equations (47), (50) to (53), and (55) are available which relate the six unknown temperatures, t_t, t_w, t_g, t_c, t_i, and t_a. For any particular room all of the shape factor terms of these equations can be evaluated from the geometry of the arrangement. The convection and conduction terms require a knowledge of the inside convective film coefficient and the conductance from each inside surface to the outside air.

Then for any room, with the outside temperature known and the panel surface temperature fixed, the inside air temperature and the average temperature of each different surface in the room can be determined by a simultaneous solution of the six independent heat balance equations.

The mean radiant temperature is then calculated from the equation,

$$\text{MRT} = \frac{(T_t^4 A_t + T_w^4 A_w + T_g^4 A_g + T_c^4 A_c + T_i^4 A_i + T_p^4 A_p)^{\frac{1}{4}}}{\text{Total area of interior surface}} - 460. \quad \ldots (56)$$

In many problems an accurate approximation can be obtained from the equation,

$$\text{MRT} = \frac{t_f A_f + t_w A_w + t_g A_g + t_c A_c + t_l A_l + t_p A_p}{\text{Total area of interior surface}} \quad \ldots (57)$$

Conditions in the room have now been established and reference to the comfort curve (Fig. 1) will determine whether or not the values of t_a and MRT which must result in order to satisfy the heat transfer relationships actually satisfy the conditions necessary for comfort (that is, fall on the curve of Fig. 1). If comfort conditions do not exist it is then necessary to vary the panel surface temperature, the panel area or the ventilation rate and repeat the analysis.

In the usual installation the panel area and ventilation rate are fixed so the panel surface temperature is the only variable subject to control. For such a case the equilibrium inside air temperature corresponding to a condition of comfort must vary with the outside temperature; in any particular installation the method of analysis indicated above makes it possible to determine t_a as a function of t_o.

All of the previous discussion has been with reference to the *average* comfort environment in a radiant conditioned room. A factor of importance which has not yet been considered is the variation in uniformity of irradiation and directional irradiation experienced by an occupant as he moves from place to place in the room. A rational approach to this problem is possible by setting up an energy balance at various positions in the room and at different points on the body surface of the occupant.

Conclusion

An attempt has been made to outline a rational method of analyzing panel heating problems and to set up general equations leading to a determination of the mean radiant and inside air temperatures corresponding to a fixed panel temperature and a known outside temperature. It has been shown:

1. That the air temperature in a radiant conditioned room is a function of the MRT and cannot be arbitrarily fixed if comfort is to be realized.
2. That the MRT can be calculated by solving simultaneously the heat balance equations for each different type of interior surface and for the ventilation air.
3. That the error likely to result from application of equations for a simple ideal system to a complex real system may be great.

As already stated, the intent of this paper is to outline the basic theory of radiant heating or cooling calculations rather than to develop methods for practical design. From the theoretical equations, the problems of practical design can be identified and their relative importance evaluated within limits. Working from the theoretical equations for the exact and general case it is possible to develop methods of expressing such necessary information as the shape factors or the series representing reflections in an actual room in terms of either simplified equations or graphs that are of direct and simple application to the problems of practical design.

The extent to which rational methods can be utilized in establishing a simple and direct practical design procedure is dependent in part on a better understanding of the physical significance of terms in the general equations but to a greater extent on the amount of experimental data available to check the accuracy and adequacy of such methods as may, from theory alone, seem to offer some promise. The authors are now at work on a comprehensive panel research program being conducted at the University of California at Berkeley as a cooperative project with the American Society of Heating and Ventilating Engineers. Progress reports will be issued at intervals.

Appendix

A) Energy leaving A_1 (of that initially emitted by A_1) and striking A_2, when e_1 and e_2 are between zero and unity (but not necessarily equal to one another). Walls perfect absorbers.

1. Energy emitted by A_1 and striking dA_2

$$dq = \frac{\sigma e_1}{\pi} \int_{A_1} \frac{T_1^4 \cos\phi_1 \cos\phi_2 \, dA_1}{r^2} \cdot dA_2 \quad \ldots\ldots (1a)$$

2. Energy reflected by A_2 and striking dA_1 (1st reflection),

$$dq_{r1} = \frac{\sigma e_1}{\pi}\left(\frac{1-e_2}{\pi}\right)\int_{A_2}\left(\frac{\cos\phi_1 \cos\phi_2}{r^2}\right)\left(\int_{A_1} \frac{T_1^4 \cos\phi_1 \cos\phi_2 \, dA_1}{r^2}\right) dA_2 \cdot dA_1 \quad \ldots\ldots (2a)$$

3. Energy reflected (2nd) by A_1 and striking dA_2,

$$dq_{r2} = \frac{\sigma e_1}{\pi}\,\frac{(1-e_2)}{\pi}\,\frac{(1-e_1)}{\pi}\int_{A_1}\left[\frac{\cos\phi_1 \cos\phi_2}{r^2}\right]\left[\int_{A_2}\left(\frac{\cos\phi_1 \cos\phi_2}{r^2}\right)\left(\int_{A_1} \frac{T_1^4 \cos\phi_1 \cos\phi_2 \, dA_1}{r^2}\right) dA_2\right] dA_1 \cdot dA_2 \quad \ldots\ldots (3a)$$

4. Energy reflected (4th) by A_1 and striking dA_2,

$$dq_{r4} = \frac{\sigma e_1}{\pi}\,\frac{(1-e_2)^2}{\pi^2}\,\frac{(1-e_1)^2}{\pi^2}\int_{A_1}\left\{\frac{\cos\phi_1 \cos\phi_2}{r^2}\right\}\left\{\int_{A_2}\left[\frac{\cos\phi_1 \cos\phi_2}{r^2}\right]\left[\begin{array}{c}\text{Second bracketted}\\ \text{term from } 3a\end{array}\right] dA_2\right\} dA_1 \cdot dA_2 \quad \ldots\ldots (4a)$$

Then the total energy leaving A_1 and striking A_2 is,

$$q = \frac{\sigma e_1}{\pi} \int_{A_2} (\text{Eq. } 1a + \text{Eq. } 3a + \text{Eq. } 4a + \ldots) dA_2 \ldots (5a)$$

$$= \frac{\sigma e_1}{\pi} \int_{A_2} (a + kb + k^2 c + k^3 d \ldots) dA_2 \ldots \ldots (6a)$$

$$\text{where, } k = \frac{(1 - e_1)\ (1 - e_2)}{\pi^2} \text{ and } a = \int_{A_1} \frac{T_1^4 \cos \phi_1 \cos \phi_2 dA_1}{r^2} \ldots \ldots (7a\text{-}8a)$$

$$b = \int_{A_1} \left[\frac{\cos \phi_1 \cos \phi_2}{r^2}\right] \left[\int_{A_2} \left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right) (\text{Term } a) dA_2\right] dA_1 \ldots (9a)$$

B) Energy leaving dA_1 and striking A_2 (same conditions as part A above).

1. Energy emitted dA_1 and striking dA_2,

$$dq = \frac{\sigma e_1 T_1^4}{\pi} \frac{\cos \phi_1 \cos \phi_2}{r^2} dA_1 dA_2 \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (10a)$$

2. Reflected (1st) by A_2 and striking dA_1,

$$dq = \frac{\sigma e_1 T_1^4}{\pi} \frac{(1 - e_2)}{\pi} \int_{A_2} \left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right) \left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right) dA_2 \cdot dA_1 \cdot dA_1 \ldots \ldots (11a)$$

3. Reflected (2nd) by A_1 and striking dA_2,

$$dq = \frac{\sigma e_1 T_1^4}{\pi} \frac{(1 - e_1)}{\pi} \frac{(1 - e_2)}{\pi} \int_{A_1} \left[\frac{\cos \phi_1 \cos \phi_2}{r^2}\right] \left[\int_{A_2} \left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right) \left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right) dA_2\right] dA_1 \cdot dA_1 \cdot dA_2 \ldots (12a)$$

4. Reflected (4th) by A_1 and striking dA_2,

$$dq = \frac{\sigma e_1 T_1^4}{\pi} \left(\frac{1 - e_1}{\pi}\right)^2 \left(\frac{1 - e_2}{\pi}\right)^2 \int_{A_1} \left\{\frac{\cos \phi_1 \cos \phi_2}{r^2}\right\} \left\{\int_{A_2} \left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right) \left(\begin{matrix}\text{Second bracket} \\ \text{from} \\ \text{Equation } (12a)\end{matrix}\right) dA_2\right\} dA_1 \cdot dA_1 \cdot dA_2 \ldots (13a)$$

The total energy from dA_1 to A_2 is then,

$$dq = \int_{A_2} (\text{Eq. } 10a + \text{Eq. } 12a + \text{Eq. } 13a + \ldots) \ldots (14a)$$

$$= \frac{\sigma e_1 T_1^4}{\pi} \int_{A_2} (a' + kb' + k^2 c' + k^3 d' + \ldots) dA_2 \cdot dA_1 \ldots (15a)$$

$$\text{where } k = \left(\frac{1 - e_1}{\pi}\right) \left(\frac{1 - e_2}{\pi}\right)$$

$$a' = \frac{\cos \phi_1 \cos \phi_2}{r^2}$$

$$b' = \int_{A_1} \left[\frac{\cos \phi_1 \cos \phi_2}{r^2}\right] \left[\int_{A_2} \left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right) (\text{Term } a') dA_2\right] dA_1$$

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Economical Air Velocities for Mechanical Air Filtration

By Frank B. Rowley* and Richard C. Jordan**

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Engineering Experiment Station, University of Minnesota.

IN the removal of impurities from the air by mechanical filters of the non-automatic cleaning type, the question often arises as to what filtration air velocity should be adopted for the most economical operation. If a comparatively high air velocity is chosen, the initial cost of the installation is reduced, as fewer filters are required to handle the necessary volume of air. On the other hand, as the resistance of the air filters to air flow varies as some power (greater than one) of the air velocity, the useful life and dust holding capacity of the filtration system between recharging or replacement of the filters may be greatly reduced. Other complicating factors involved in the determination of the most economical velocity for air filtration, in addition to the cost of replacement or recharging of the filters and the yearly fixed charges or depreciation on the initial installation, are filtration power costs and the rental value of space occupied by the system. Analysis may be complicated further by variations in efficiencies of filtration with changes in air face velocity, although tests conducted[1] over a range of practical air velocities indicated little variation in efficiency when determined on a weight basis.

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[1]Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed and Air Velocity through Filter, by F. B. Rowley and R. C. Jordan. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 415.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, San Francisco, Calif., June, 1941.

Velocity-Resistance Relationship

One of the important variables affecting the economy of air filtration is the relationship between air velocity and filter pressure drop. Theoretically, if the air flow through the filter is completely turbulent, the resistance to flow varies as the square of the velocity, while if the flow is laminar or stream-lined, the resistance varies directly as the first power of the velocity. Because of the nature of the media through which the air is passing, it would appear likely that in the case of mechanical air filters a combination of both laminar and turbulent flow might occur.

In order to determine the actual relationship between air velocity and filtration energy losses, four typical filters were chosen for experimental laboratory tests. These filters may be described briefly as follows:

A—A permanent type of cleanable oil filter, 4 in. thick, with 24 layers of expanded metal and wire screen graded from coarse mesh at entrance to fine mesh at leaving side.

B—A viscous coated throw-away type filter 2 in. thick. The fibrous media were graded in fiber size, density, and oiling from entering to leaving side.

C—A cellular type filter 2 in. thick built in 2 sections, each with the axis of cells set at 45 deg to the center line of duct and at 90 deg to each other. The cells on the entering side were of larger dimensions than those on the leaving side of the filter.

D—A filter of cotton media of coarse material on the entering side and glazed on the leaving side. The filter media were accordion plaited in frame to give an area of approximately 12 times the cross sectional area of the air stream.

Each of these filters was subjected to face air velocities of 200, 300, 400 and 500 fpm under conditions of no dust loading and at dust loadings corresponding to approximately 0.2, 0.3, 0.4, and 0.5 in. of water filter pressure drop at a 300 fpm face air velocity. These velocity-resistance tests were made at various conditions of dust loading in order to determine what changes, if any, were effected by differences in the dust loading pattern. Because of the accordion plaited arrangement of the filtering media in Filter *D*, these face air velocities (as based upon the projected filter area) were equivalent to media face velocities of only 16, 24, 33, and 41 fpm. The filter test apparatus used for determining the air velocities and the filter energy losses, and for feeding the dust to the filters has been described in previous publications.[2,3] The dust used in loading the filters was fed at a rate of 40 grams per hour and consisted of 50 per cent by weight of Pocahontas ash, 20 per cent Illinois fly ash, and 10 per cent Fuller's earth, all screened through a 200 mesh screen, and 20 per cent lampblack screened through a 100 mesh screen.

As a result of these tests it was found that the relationship between velocity and filter resistance could be expressed most accurately by the equation,

$$R = K V^n + C \quad \ldots\ldots\ldots\ldots (1)$$

where, R = filter resistance, inches of water

V = face air velocity, feet per minute

[2]Loc. Cit. Note 1.

[3]Factors Affecting the Performance and Rating of Air Filters, by F. B. Rowley and R. C. Jordan. (Bulletin of the Engineering Experiment Station No. 16, November, 1939, University of Minnesota.)

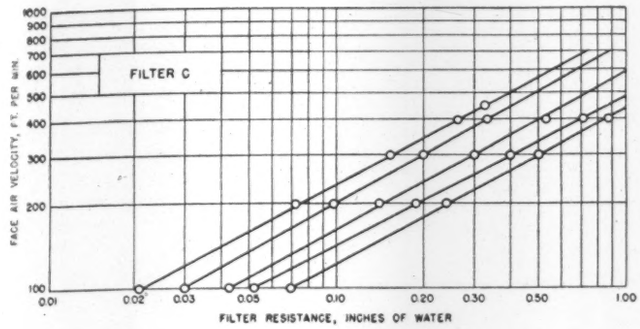

Fig. 1—Velocity-resistance curves for Filter *C* under different conditions of dust loading

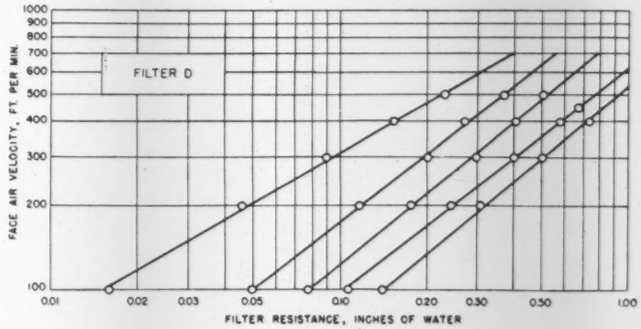

Fig. 2—Velocity-resistance curves for Filter *D* under different conditions of dust loading

C, K, n = constants, depending on filter design and loading.

However, the constant C was in all cases less than 0.01 in. of water, and, therefore, a close approximation to the actual conditions can be determined by a re-evaluation of the constants, K and n, and the relationship

$$R = K V^n \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Figs. 1 and 2 show log-log graphs of the velocity-resistance test data for Filters *C* and *D*. Table 1 shows a summary of the velocity-resistance relationships for all filters tested as obtained from these and similar graphs. The exponent, n, is determined from the slope of the graph; the constant, K, varies with both the velocity and the dust loading and has not been recorded as it has little practical significance. The closest approach to completely turbulent flow is found with Filter *C*, in which the resistance varies as the 1.83 power of the velocity. The closest approach to laminar flow is found with Filter *D*, in which the resistance of the filter, other than for the initial unloaded condition, varies as the 1.21 power of the velocity. It is interesting to note that with this filter the velocity-resistance relationship is different for the initial unloaded condition than it is when containing some dust load. Apparently the flow is more nearly turbulent when unloaded than when loaded. All of these tests indicate that for unit mechanical filters of usual design the air flow can be classed as neither completely laminar nor completely turbulent, but probably a combination of the two. It is probable that the conditions of flow change in different sections of the filters because of the gradients in media size and pack density. The range of velocities used in these tests was limited to approximately that encountered in practice and it is probable that if this range had been extended, marked variations in the velocity exponents would have been found.

Effect of Velocity on Filter Life and Dust Holding Capacity

In order to illustrate the effect of face air velocity on filter life and dust holding capacity a series of standard performance tests was run on all four filters by the weight or crucible method[4] using the 50-20-20-10 dust mixture mentioned earlier at a rate of dust feed of 40 grams per hour and a face air velocity of 300 fpm. The resistance curves thus obtained, along with the relationships of Table 1, were used to determine the resistance curves for face air velocities of 200, 400 and 500 fpm. Figs. 3 and 4 show typical families of resistance curves obtained for Filters *C* and *D* in this manner.

These resistance curves may be used as a basis for determining the effect of face air velocity on filter life and dust holding capacity if an arbitrary limiting resistance is set beyond which it may be considered uneconomical to operate the filter. This is difficult to do, however, as such a limiting resistance depends upon a variety of conditions. With the majority of installations 0.4 in. of water is usually considered as a reasonable limit with a face air velocity of 300 fpm, but this does not necessarily apply to other velocities. In general, after a sufficiently heavy deposit of dust has been collected in the filter, the resistance curve tends to rise so rapidly that increasing the allowable maximum resistance has comparatively little effect in lengthening the life and dust holding capacity.

Two methods of determining filter life and dust holding capacity were used in the present calculations. In one case the limiting resistance was taken as 0.4 in. of water, regardless of face air velocity; in the second, the life was determined by limiting the total resistance rise across the filter to 0.3 in. of water. The results of these calculations are shown in the first seven columns of Table 2. In interpreting these results it must be remembered that the rate of feed was kept constant regardless of changes in air velocity, whereas in actual filtering installations increasing the face air velocity results in a proportional increase in the amount of dust fed to the filter. The results of these tests indicate a marked increase in filter life and dust holding capacity with decrease in air velocity. As would be expected, the greatest increases in performance are found in all cases when the limiting resistance is taken as 0.4 in. of water rather than when based on a 0.3 in. of water resistance rise. This is caused by the tendency for the initial filter resistance to approach and eventually to equal the limiting resistance as the velocity increases. This face velocity at

Table 1 — Velocity-Resistance Relationships for Filters *A*, *B*, *C*, and *D*

Filter	Condition of Filter	Velocity-Resistance Relationship
A	Both unloaded and with all dust loads	R varies as $V^{1.73}$
B	Both unloaded and with all dust loads	R varies as $V^{1.60}$
C	Both unloaded and with all dust loads	R varies as $V^{1.83}$
D	Unloaded	R varies as $V^{1.67}$
D	With all dust loads	R varies as $V^{1.71}$

[4] ASHVE Standard Code for Testing and Rating Air Cleaning Devices used in General Ventilation Work. (ASHVE Transactions, Vol. 39, 1933, p. 225.)

which the limiting and initial resistances are equal, and consequently the life zero, is 436 fpm for Filter *A*, 575 fpm for Filter *B*, 537 fpm for Filter *C* and 1028 fpm for Filter *D*.

The ratios of the life in hours at 200 fpm to the life at 500 fpm when based upon a 0.3 in. of water resistance rise for Filters *A*, *B*, *C*, and *D* respectively, are 1.4, 2.7, 2.7, and 1.5. The same ratios when based upon a final resistance of 0.4 in. of water are in the same order, infinity, 5.7, 8.0, and 1.8. This last set of ratios is somewhat misleading as it is unlikely that a 0.4 in. of water final resistance would be chosen in combination with a 500 fpm air velocity. However, a 400 fpm face air velocity in combination with this limiting resistance is quite possible, and certainly the first set of ratios based on a 0.3 in. of water resistance rise for all velocities and filters is a fair basis for comparison.

In order to obtain a rational basis for translating these laboratory values of filter life to actual filter life values a series of five tests was run to determine the normal weight concentrations of dust contained in air. These determinations were made by drawing samples of laboratory air through alundum crucibles and measuring the increases in crucible weight. The rate of sampling was 1 cfm and the time of sampling 300 min. The air-dust concentrations as determined by these tests ranged from 0.1175 to 0.0664 grains per 1,000 cu ft with an average value of 0.0845. Approximately this same average concentration was obtained as the result of an additional series of tests in which the concentrations were determined by electrical precipitation of the dust onto aluminum foil surfaces. All of these values were determined during periods when there were no unusual activities occurring in the laboratory which might cause abnormal dust concentrations and may be considered as representative of the concentrations to be found in typical urban areas.

The last two columns of Table 2 show the actual filter life in hours, based upon an air-dust concentration of 0.0845 grains per 1,000 cu ft and for both methods of determining limiting filter resistance. These values have been calculated by multiplying the laboratory filter life in hours by the ratio of the laboratory rate of dust feed to the rate of dust feed resulting from the passage of normal air through the filter. This latter rate of feed varies with the air velocity as it is assumed that in this case the weight of dust per unit volume of air remains constant. Therefore, it should be noted that, in order to remove the same amount of dust with an air velocity of 200 fpm as in the case of a 500 fpm air velocity, it is necessary to have 2½ times as much filtering area, as the rate of filtration is only 40 per cent as great. Whereas these values for actual filter life may be considered as reasonable approximations to what may be expected in practice, it should be remembered that values of both laboratory and actual filter life and dust holding capacity will vary widely with the nature of the test dust, and in these determinations it has been assumed that both the artificial laboratory test dust and the normal atmospheric dust suspensions have approximately the same physical characteristics.

Table 3 shows the variation of average filter resistance with face air velocity when based upon both a

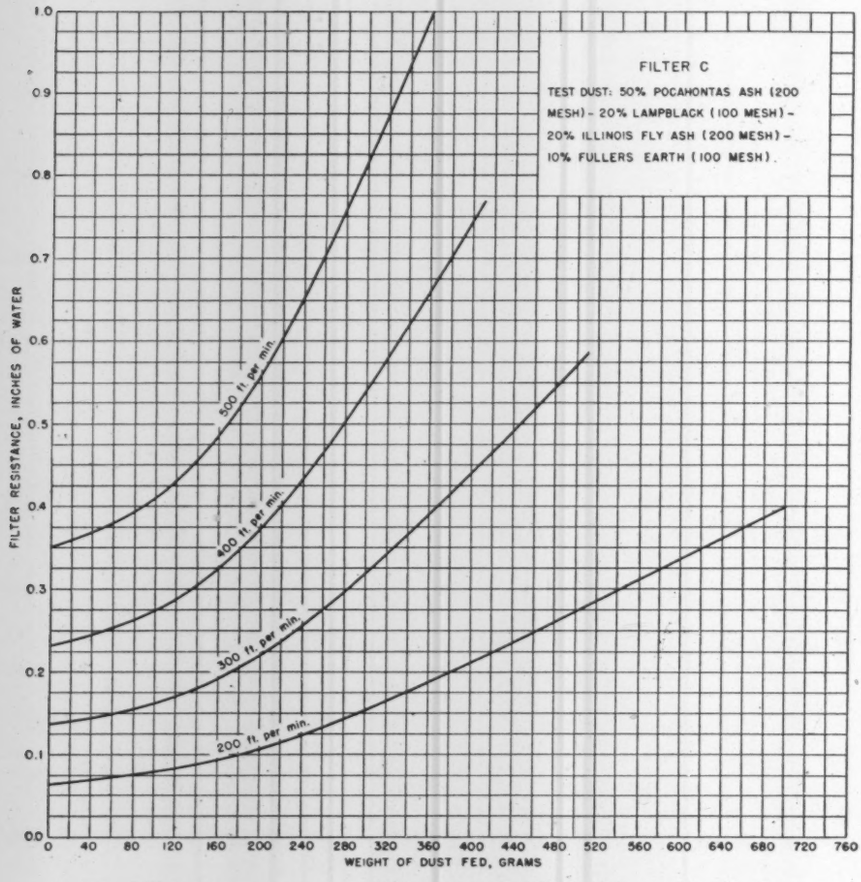
FILTER C
TEST DUST: 50% POCAHONTAS ASH (200 MESH) - 20% LAMPBLACK (100 MESH) - 20% ILLINOIS FLY ASH (200 MESH) - 10% FULLERS EARTH (100 MESH).
FILTER RESISTANCE, INCHES OF WATER
WEIGHT OF DUST FED, GRAMS
500 ft. per min.
400 ft. per min.
300 ft. per min.
200 ft. per min.

Fig. 3—Resistance curves for Filter *C* with different face air velocities
(40 g per hour rate of dust feed)

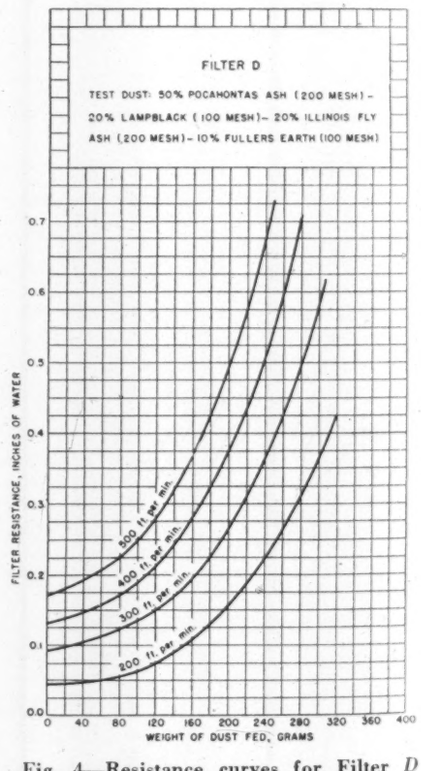
FILTER D
TEST DUST: 50% POCAHONTAS ASH (200 MESH) - 20% LAMPBLACK (100 MESH) - 20% ILLINOIS FLY ASH (200 MESH) - 10% FULLERS EARTH (100 MESH)
FILTER RESISTANCE, INCHES OF WATER
WEIGHT OF DUST FED, GRAMS
500 ft. per min.
400 ft. per min.
300 ft. per min.
200 ft. per min.

Fig. 4—Resistance curves for Filter *D* with different face air velocities
(40 g per hour rate of dust feed)

Table 2—Effect of Air Velocity on Life and Dust Holding Capacity for Filters *A*, *B*, *C*, and *D*
(*Laboratory Test Dust Feed* 40 *g per hour. Dust mixture* 50, 20, 20, 10)

Filter	Average Efficiency, Per Cent	Face Air Velocity, Ft/Min	Filter Life by Laboratory Test, Hours		Dust Holding Capacity by Laboratory Test, Grams		Rate of Dust Feed to Filter with Air Containing 0.0845 Grains per 1000 cu ft, Grams per Hour	Filter Life with Air Containing 0.0845 Grains Dust per 1000 cu ft, Hours	
			0.4 in. H_2O Final Resistance	0.3 in. H_2O Resistance Rise	0.4 in. H_2O Final Resistance	0.3 in. H_2O Resistance Rise		0.4 in. H_2O Final Resistance	0.3 in. H_2O Resistance Rise
A	80.3	200	19.0	19.0	610	610	0.183	4150	4150
		300	15.3	16.8	491	540	0.274	2235	2455
		400	7.0	15.2	225	488	0.366	765	1660
		500	0.0	13.8	0	444	0.457	0	1210
B	81.7	200	17.2	16.1	563	526	0.183	3750	3510
		300	9.2	10.1	301	330	0.274	1340	1470
		400	5.5	7.5	180	245	0.366	602	820
		500	3.0	6.1	98	199	0.457	263	534
C	69.9	200	17.5	16.2	490	454	0.183	3830	3540
		300	9.2	9.9	257	277	0.274	1350	1450
		400	5.5	7.4	154	207	0.366	601	809
		500	2.2	6.0	62	168	0.457	193	529
D	93.5	200	7.8	7.3	292	273	0.183	1710	1600
		300	6.3	6.2	236	232	0.274	921	905
		400	5.2	5.5	195	206	0.366	570	603
		500	4.3	4.9	161	183	0.457	378	430

final limiting resistance of 0.4 in. of water and also a limiting resistance rise of 0.3 in. of water. These average resistance values were determined by finding the areas under the resistance curves (see Figs. 3 and 4) by means of a planimeter and then dividing these values by the abscissae. In all cases the average resistance increases rapidly with increased air velocity, and in general the average resistance at a 500 fpm face air velocity is approximately two to three times that at a 200 fpm face velocity. The air horsepower required to overcome the filter resistance is directly proportional to the average resistance, and therefore, power costs resulting from the use of high air velocities may become an important item.

Derivation of Filtration Cost Equation

It is apparent from the preceding analysis that the correct choice of face air velocity is important for economical filtration because of its effect on filter life and average resistance. However, in order to determine the most economical air velocity, it is necessary to consider concomitantly all cost factors which are affected by changes in air velocity. With this end in view, let

Q = rate of filtering, cfm
t = time of filtering, hours per year
C = initial cost each filter (including installation labor charges) or cost of reconditioning, dollars
V = filter face air velocity, feet per minute
L = life of filter, hours
R_a = average resistance of filter, inches of water
H = average air horsepower[5] required for filtration $= \dfrac{R_aQ}{6350}$
D = power cost, dollars per air horsepower-hour
A = filter area, square feet
I = total annual fixed charges or depreciation, per cent
B = cost original filtering installation, dollars per square foot filter area
F = rental value of space occupied by filters, dollars per square foot filter area per year

then

Volume of air to be filtered per year, cubic feet $= 60Qt$

Volume of air handled by each filter, cubic feet $= 60AVL$

Number filters (new or re-conditioned) required per year $= \dfrac{Qt}{AVL}$

Cost of filters (new or re-conditioned), dollars per year $= \dfrac{Qtc}{AVL}$

Power cost, dollars per year $= HtD = \dfrac{R_aQtD}{6350}$

Fixed charges, dollars per year $= \dfrac{QBI}{100V}$

Rental value of occupied space, dollars per year $= \dfrac{QF}{V}$

The total cost of air filtration per year will then be the sum of these individual cost items, or

Yearly cost air filtration

$$= \frac{QtC}{AVL} + \frac{R_aQtD}{6350} + \frac{QBI}{100V} + \frac{QF}{V} \quad (3)$$

In analyzing Equation (3) it is apparent that both R_a, the average resistance to air flow, and L, the filter life, are variables dependent upon air velocity. In a study of filtration cost in any specific instance, it is, therefore, necessary to evaluate these in terms of V, the air velocity.

In practically all instances, filter life, L, is an inverse function of V raised to some power greater than 1. Therefore, despite the fact that V appears in the denominator, the expression for cost of new or reconditioned filters per year generally increases as the velocity increases. As the average resistance, R_a, varies directly as some power of V, the yearly power costs will also increase as V increases. The yearly depreciation and space charges both decrease with increased velocity as both of

Table 3—Effect of Air Velocity on Average Resistance For Filters *A*, *B*, *C*, and *D*
(*Dust Feed* 40 *grams per hour. Dust mixture* 50, 20, 20, 10)

Filter	Average Resistance, in. of Water (0.4 in. Water Final Resistance)				Average Resistance, in. of Water (0.3 in. Water Limiting Resistance Rise)			
	Face Air Velocity, ft/min				Face Air Velocity, ft/min			
	200	300	400	500	200	300	400	500
A	0.15	0.26	0.36	.....	0.16	0.28	0.43	0.61
B	0.20	0.23	0.28	0.35	0.20	0.25	0.34	0.43
C	0.20	0.23	0.29	0.36	0.18	0.25	0.35	0.47
D	0.14	0.18	0.22	0.25	0.13	0.18	0.23	0.27

[5]See any treatise on fan engineering for derivation.

these expressions contain V only in the denominator.

In determining filtration costs under practical operating conditions by means of Equation (3), there is in every case an optimum air velocity which results in minimum yearly filtration charges. This velocity may be determined by differentiating the yearly cost expression with respect to V, setting the resulting expression equal to zero and solving for V. However, the solution is, in most cases, quite difficult because of the number of terms in the original expression, and it is, therefore, usually much simpler to assume four or five velocities, determine the corresponding yearly filtration charges, and then determine the velocity corresponding to minimum cost graphically.

It must be realized that analytical methods such as are made possible by the application of Equation (3) have definite practical limitations because of the difficulties encountered in the evaluation of some of the terms. Usually an approximate numerical expression must suffice. However, calculation of the optimum velocity for each case encountered is not always necessary, as once the weight to be placed on each of the factors has been established by the analysis of a few cases, general engineering judgment will usually dictate velocities reasonably close to the optimum. The following analysis based upon actual data for a typical installation is presented as an illustration of the procedure to be followed in determining the optimum air filtration velocity.

Example: It is desired to determine the optimum filter face air velocity for an air filtration system to be operated in conjunction with the ventilation system of a building designed to handle 10,000 cfm for an average of 3,000 hours of operation each year. Unit, disposable filters, each 20 in. by 20 in., are to be used. The replacement cost of these filters, including labor charges, is $1.25 each. Power is available at $0.02 per kilowatt hour; or, assuming a fan efficiency of 70 per cent and a motor efficiency of 90 per cent, the power costs are $0.024 per air horsepower-hour. The total annual fixed charges on the installation, including interest charges on the capital expenditure and annual depreciation, are to be taken at 12 per cent.

In order to evaluate the yearly fixed charges and the rental value of the occupied space it is first necessary to determine the relationship between the number of filters and the cost of the necessary duct work and filter frames for the installation and the relationship between the number of filters and the floor area necessary for the installation. In the solution of the present problem the ductwork for several sizes of installations was drawn up, estimates were obtained on the cost of this ductwork, and the floor area required for these installations measured. These relationships were thus found to be roughly linear. For example, in the case of the initial installation cost, it was found that in taking the number of filters in the installation at 9, 18, 27, and 36 the minimum cost per square foot of filter area was $3.71 and the maximum $4.32. Therefore, the initial cost of the duct work and filter frames may be taken as approximately $4.00 per square foot of filter face area. In the same way the rental value of the space occupied by the additional equipment (duct work, filter frames, etc.) required for the filtration system was determined and may be taken as approximately $1.90 per year per square foot of filter face area. These figures are based upon the prevailing costs for the particular locality under consideration.

In order to evaluate the power costs and the cost of replacement filters it is first necessary to determine the relationship between actual life in hours, L, (based on a limiting resistance of 0.4 in. of water and a dust concentration 0.0845 grains per 1000 cu ft air) and face air velocity, V, and the relationship between average resistance, R_a, and air velocity, V. Actual laboratory tests (similar to those shown in Table 2) were made on the particular filter chosen for installation and these relationships found to be

$$L = \frac{(600 - V)^{2.04}}{1972}$$

and

$$R_a = \frac{V^{0.704}}{227}$$

In the solution of this problem, face air velocities of 100, 200, 300, 400, and 500 fpm were chosen and the yearly filtration costs calculated by substitution of the given conditions in Equation (3). The graphical solution, with these air face velocities plotted against the calculated total yearly filtration costs, is shown in Fig. 5. This curve indicates that the optimum air velocity, as far as filtration costs are concerned, is in the neighborhood of 300 fpm. In this particular case, any air velocity chosen between 200 and 400 fpm would be reasonably satisfactory. However, below 200 fpm and above 400 fpm, the air filtration costs rise very rapidly and at 100 fpm are approximately double and at 500 fpm almost triple the minimum cost. If filter life had been determined for a limiting resistance rise of 0.3 in. of water instead of the basis used, the yearly filtration costs would not have increased quite as rapidly with the higher air velocities.

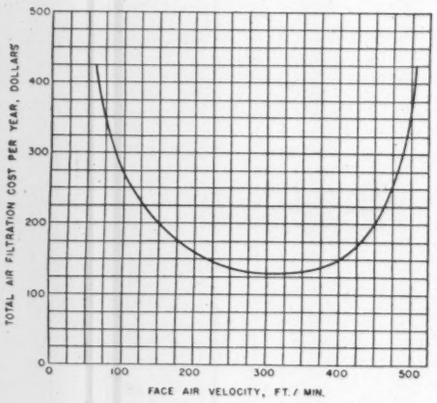

Fig. 5—Graphical solution of example showing variation of yearly filtration costs with face air velocity. Rate of filtration 10,000 cfm

Summary

In this paper it has been shown that the relationship between filter resistance, R, and the face air velocity, V, for mechanical filters is given by the equation $R = KV^n$ where K and n are constants and V is limited to practical operating velocities. The exponent n is dependent upon the design of the filter and in some cases upon the filter loading. For all filters tested the values of n were between 1 and 2 indicating that the air flow through the filters was neither completely laminar nor completely turbulent, but probably a combination of the two.

It has been shown that the face air velocity chosen in the operation

of a filter has a marked effect upon the life and dust holding capacity of the filter when these performance characteristics are based on either a final limiting resistance or upon an allowable resistance rise. When the laboratory values of filter life are translated into actual values of filter life, the effect of air velocity is even more marked, since under practical operating conditions decreasing the air velocity decreases also the rate of dust feed to the filter.

The total yearly cost of air filtration has been shown to be dependent upon several factors each of which varies to some degree with the face air velocity. An equation has been derived showing the relationship between air filtration costs and air velocity. The procedure has been outlined by which it is possible to determine the optimum air velocity for minimum filtration costs.

NEW OFFICERS AT NORTH CAROLINA

May 2, 1941. The North Carolina Chapter of the Society held its annual meeting at Page Hall, North Carolina State College, Raleigh, for the specific task of electing their new officers, who are as follows:

President—W. M. Wallace, II
Vice-President—H. H. Hill
Secretary-Treasurer—F. J. Reed.
Board of Governors—W. M. Wallace, II, H. H. Hill, F. J. Reed, Arvin Page, C. Z. Adams and L. L. Vaughan.

The meeting was also for the transaction of the usual business along with the election of the officers, and before the meeting adjourned it was announced that a summer meeting is being planned for Pinehurst and details will be announced later.

SOUTHERN CALIFORNIA ELECTS OFFICERS

May 14, 1941. The May meeting of the Southern California Chapter was held in Los Angeles, and Henry Sachs, chief engineer, Utility Fan Corp., gave a very interesting talk on evaporative cooling. He gave the history, problems and application of the evaporative cooler, very often referred to in the western part of the United States as *desert coolers*, and his talk was received by the audience with considerable interest.

There were 27 members and guests in attendance, and the following officers were elected to serve for the coming season:

President—A. J. Hess
Vice-President—H. H. Bullock.
Secretary—Leo Hungerford
Treasurer—W. O. Stewart
Board of Governors—H. H. Douglas, J. Frank Park, Maron Kennedy and J. B. Griffith.

ASTM HOLDS MEETING

Intensive interest in the field of engineering materials, particularly in the development of knowledge of materials, and in the standardization of specifications and tests, was evident at the 44th annual meeting of the *American Society for Testing Materials* held at the Palmer House, Chicago, Ill., during the week of June 23. During this same week the 6th Exhibit of Testing Apparatus was in progress, stressing the importance of satisfactory machines, laboratory equipment, and supplies in testing and insuring quality of various products.

The registration of members, committee members and guests exceeded that of any other *ASTM* meeting, reaching a total of 1553.

Progress in the field of standardization was indicated by the 70 new specifications and tests published for the first time, many providing quality requirements and procedures for testing important engineering materials and products.

At the technical sessions there were 115 technical papers and reports presented. It is reported that all of the sessions and committee meetings were well attended.

E. M. JOLLEY DIES

The many friends of Edward McKinley Jolley will be shocked to learn of his sudden passing on July 3, of a heart attack while in his car on U. S. highway No. 10, three miles east of De Soto, Kans.

Mr. Jolley was born May 23, 1905, at Spearfish, S. D., and attended the Spearfish State Normal School in preparation for his studies at the University of Nebraska, from which he received his M.E. and B.S. in 1928. He was a member of the Pi Kappa Alpha fraternity.

From 1928 to 1937 he was sales engineer for the Worthington Pump and Machinery Corp., assigned to the Pittsburgh, Pa., Harrison, N. J., and Kansas City offices, from which he resigned in 1937. At that time he joined the staff of The Marley Co., Kansas City, Mo., as sales engineer, which position he held at the time of his death. Mr. Jolley was a familiar caller on most of the firms engaged in summer air conditioning and refrigerating work.

Mr. Jolley joined the ASHVE in 1938 and served as secretary of the Kansas City Chapter during the past year, and was recently elected to vice-president of the Chapter. He was also active in the Engineers' Club of Kansas City, and in the affairs of the *ASME* and *ASRE*. He

E. M. Jolley

will be remembered by those who attended the Society's 47th Annual Meeting in Kansas City last January, as he was a member of the Committee on Arrangements and was chairman of the Transportation Committee.

Funeral services were held at 4:30 p. m. on Saturday, July 5, at the Freeman Chapel, 42nd St. and Baltimore Avenue, Kansas City.

Mr. Jolley made his home at 11 West 73rd St. Terrace, and is survived by his widow, Mrs. Katherine Hinckley Jolley, and two young daughters, aged 7 and 5, to whom the Officers and Council of the American Society of Heating and Ventilating Engineers extend their sincerest sympathy. He also leaves his mother, Mrs. Thomas F. Jolley, Laurel, Mont., a twin brother, Theodore, Butte, Mont., and two other brothers, Joseph of Laurel and Raymond of Chicago.

The passing of "Ed" Jolley will be keenly felt by the host of friends he had developed through service and a remarkably winning personality. Extremely devoted to his work, his friends and his family, an outstanding able engineer and a stellar salesman, he was received everywhere with a warmth that reflected the genuine admiration accorded him by all with whom he came in contact.

Proceedings of the Semi-Annual Meeting 1941

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Palace Hotel—June 16-20, 1941—San Francisco

Pres. J. E. McNevin (*HPACCNA*) and Pres. W. L. Fleisher (ASHVE) are greeted by Mayor Angelo J. Rossi

SAN FRANCISCO was the scene of the 1941 Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the 52nd Annual Convention of the *Heating, Piping and Air Conditioning Contractors National Association,* during the week of June 16. Because this was the first occasion on which either of the organizations had met on the Pacific Coast it was fitting to have the opening session as a joint function, after which the membership of both organizations attended the opening of the Pacific Heating and Air Conditioning Exposition.

Headquarters for the ASHVE was established at the Palace Hotel, and 360 members and guests registered. The St. Francis Hotel was the center of activity for the *HPACCNA* and their registration numbered 450.

JOINT SESSION—

Monday, June 16, 10:00 A.M.

The joint session of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *Heating, Piping and Air Conditioning Contractors National Association* was called to order by Walter L. Fleisher, president of the ASHVE, at 10:00 a. m., June 16, in the Colonial Room of the St. Francis Hotel, San Francisco, Calif.

Major Harry A. Shuder, U. S. Army Chaplain gave the invocation and John Henderson sang "God Bless America." The audience then sang "America," led by Messrs. Henderson and N. H. Peterson.

Chairman Fleisher introduced the Honorable Paul Peek, secretary of State of California, who gave a message of welcome on behalf of Governor Olson, and a brief response was made by J. E. McNevin, president of the *HPACCNA.* Clyde E. Bentley, president of Golden Gate Chapter of the ASHVE extended a welcome on behalf of the local chapter and Daniel Hayes, president of the *Heating, Piping and Air Conditioning Contractors of San Francisco,* assured the members assembled that it was a great pleasure for the local contractors to entertain the visiting members.

Mr. McNevin presented Frank E. Moran, president of the *Chicago Master Steamfitters Association.* Mr. Moran said that the Chicago association for many years had enjoyed a very conservative and well directed guiding hand centered in Louis Braun, who has just completed his 25th year as executive secretary. In commemoration of this occasion, he said, "the Chicago association desires to present a testimonial of appreciation, and, as president, I take great pleasure in presenting this diamond ring to Mr. Braun."

Mr. Braun made a brief response and expressed his appreciation to the Chicago members for their fine tribute to his service.

Chairman Fleisher then called on G. M. Simonson, chairman of the ASHVE Committee on Arrangements, to give some announcements about the entertainment.

Chairman Fleisher introduced Mr. A. Appleton, president of the San Francisco Chapter of *American Institute of Architects,* who gave an entertaining discussion on the necessity for close collaboration between architects, engineers and contractors. Mr. Appleton spoke in part as follows:

Mr. President and officers and members and their friends and my friends and ladies of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS meeting here at its Semi-Annual Meeting, and the *Heating, Piping and Air Conditioning Contractors National Association* assembled for its 52nd annual convention, in joint meeting: greetings and welcome from the architects of California.

For myself, it is a happy coincidence, that at this time when you are gathered here, I hold office in the Northern California Chapter of the *American Institute of Architects* and, therefore, as its president may I extend to you special and additional greetings and welcome from our Institute.

We architects are honored that you have invited one of us to be with you this day. It is my hope that I might fulfill, if only in a small measure, the responsibility involved by the acceptance of that invitation.

It is unusual for an architect to come before a large group of engineers, technicians and contractors such as yourselves.

Engineers, in the minds of the architects, are a very serious group of individuals. They do take themselves and their work very, very seriously. I would like to see them relax more.

The work and practice of the architect today requires that he keep in constant touch with his collaborating engineers. The development and changes in those branches of our industry with

which you are concerned keeps us forever alert and our work flexible and fluid to adopt the new and better things that you provide for us, and which we distribute to the public through our works.

It would be very interesting to picture the average American dwelling place of half a century ago; the cooking element serving as the central heating plant and the sanitary conveniences of primitive and uncertain character, and compare these with the standard equipment of the average home of today, which equipment was then not available and which only gradually through the years became obtainable by the average family and that of the upper income brackets.

A great percentage of our construction has always been in the residential class. Untold thought, unlimited energy, uncounted hundreds of dollars have been invested in devising and designing that average home which today so many American families enjoy.

Our countryside overflows with new home developments and no matter what anyone or any group might say to the contrary, these home units are truly miracles. It is a far journey from the hand-pumped water supply in the backyard to the faucet convenience in the kitchen of today. It is a far journey from a hot-water supply in the kitchen stove to the instantaneous and voluminous supply of today through the many hot-water heating devices. It is a far journey from the privy near the back fence to the streamlined sanitary conveniences of the bathroom of the average American family of today.

I would like to dwell for a moment on this average American home of today. It is of inescapable importance in the well being of our nation. The well being of a nation is measured in the well being of its people.

There are many of you, perhaps, who have not had cause or reason to explore, to investigate that new average American home. You have seen thousands of them spreading out in new land developments, and subdivisions around all of our large cities. Small, attractive, compact, streamlined and efficient gems of construction and enterprise.

And, gentlemen, you have played no inconsiderable part in accomplishing much of this. Within the reach of practically all one can have today those things which were enjoyed only by a few a short time ago, and by no one just a few years before then. A kitchen well lighted with the new utilities for cooking, refrigerating and ventilation is the usual thing. The bath room is equipped with fixtures and fittings in style and quality almost beyond belief, and forever being improved upon, and all contributing to the personal physical and social welfare of our country.

The architects' task of yesterday was a relatively simple and easy one. Requirements and standards were simple and fixed. The architects' task of today is a complex one. Requirements and standards are complex and changing. The architects' task of tomorrow? Who knows?

We do know that the apparatus we employ today will not equal that offered us tomorrow. The very nature of the practice of the architect has been changed by the elements he can and must employ. As a matter of fact, the architect is a coordinator. He gathers together a thousand parts, places and arranges these in such a manner to form a properly functioning whole. His responsibility increases as he uses the products of your imagination and manufacture.

Who of us would have the temerity to undertake the planning of a building and the specifying of the parts that go into that building without the collaboration of you engineers? The architect must, and he owes it to the employer to so do.

There are organizations today manufacturing mechanical apparatus which they offer to the architect and guarantee the proper functioning thereof. The architect cannot and must not assume these things without the use of independent and unbiased experts. The allure to save an engineer's fee must be discarded.

The comfort the architect finds in referring to his engineer any faults or weaknesses in the mechanical plant of his building is in itself worth the entire fee paid to such engineer.

Although I may have devoted too much effort discussing the residential construction and aspect of our business, that which applies there in truth applies to these other types of practice that the architect enjoys; the other part of his practice consisting of commercial and industrial works. Perhaps, at no other time in our history has the talent of the building industry been called upon to undertake the gigantic task that is facing it today.

There is at present functioning within San Francisco, a city very vulnerable in the event war comes to us out of the Pacific, a large group of men composed of a cross-section of the entire building industry with a good percentage of engineers and architects participating therein.

You must all prepare to undertake a part of the job of defense.

ASHVE and *HPACCNA* in Joint Session at San Francisco

The vital and important part your British colleagues are now playing is contained within a report that I have before me.

It is not my intention to read this report, but it is of great and terrible interest. It is entitled "National Technological, Civil Production Committee" appointed by the Secretary of War. It is authentic information secured in Britain by our American experts.

A good portion of this report pertains to those matters in which you gentlemen qualify as experts. I note that the very first two and probably most important sections are devoted to sanitary and public health and power.

Under heading number one and listed as Subjects A, B, C and D are the titles, "Sewers, water supply, refuse and garbage disposal, public health."

Under number two, under "power" listed under A, B, C and D, "Electric, gas, communications," there is outlined a terrifying situation that is being met in a heroic way by the British engineers. It is horrifying to read this report and see the devastation that occurs under the situation with which they are continually confronted.

The job the engineers have undertaken in Britain has been a terrible one, but it has been excellently and magnificently met. This report here tells what happens when certain instruments of destruction fall upon the cities of Britain and what might be expected, and in a cold, hard matter of fact way it indicates the solution to each one of those problems, and we can learn from this and prepare ourselves from this.

I understand that one of the great problems facing our British friends when they are forced to seek shelter and protection underground is that concerned with the air supply, and strange as it seems there is not a great deal mentioned herein.

I am told that air conditioning has been the saving factor and that one of our very large companies specializing in this particular branch of work is doing a wonderful job. With this and other engineering and mechanical aids that are being employed it might well be believed that the control switches in the hands of the British engineers will prove to be as vitally important a force in the victory that will ultimately come as the switches in the hands of the British fighting forces controlling the guns.

This air conditioning which was given us but yesterday in a practical and economical form, is today accepted and employed as a matter of course. Our clients with commercial and industrial projects assume that their buildings will be equipped with adequate machinery for air conditioning.

I dwell on this because, like all of you, I am interested in the well-being of my fellow man. This great improvement has added to that well-being, and his well-being, the industrialist has learned, and has had proven to him that it is good business for himself.

The world crisis which has fired our own national crisis has affected our entire building industry, private and public. The usual program of employing an architect to plan and specify and to make comparative bids in many instances has given way to the placing of tremendous contracts directly into the hands of the contractor, he to employ his own architects and engineers, and thereby practically determining the type of construction and professional fees to be paid.

The Government states that speed is the essence, but, in my humble opinion, I believe that with the employment of those engineering and architectural offices and organizations now in existence with years of experience behind them, they can do the job the Government requires, and on the tried and accepted basis of taking bids and awarding the contract to the lowest bidder.

The practice of Government has resulted in concentrating in the hands of comparatively few, unbelievable and tremendous amounts of building construction.

Now ladies and gentlemen, I have adequately used the moments given me and in closing I want to forget that I am an architect and that you are engineers. I want to close by having you join me in the thought that we are all Americans faced with a tremendous task today and it is at such meetings as this where we do collect, that we can tell these things to each other. We must look forward to the future with a stern face and with a sound grip on our hearts. I cannot but again welcome you to California and welcome you to our fair city, and when the winds go down and the fog rolls away you will see vistas of grandeur that you will remember forever.

Chairman Fleisher announced that the official convention photograph would be taken after which Thomas Douglass of San Francisco made some announcements about the entertainment program which were of interest. Announcement was also made by Mr. McNevin of the appointment of the *HPACCNA* credentials committee.

Chairman Fleisher said that in his opinion the joint session should not close without having a few remarks from some of the men who have been instrumental in the building of both organizations. He then introduced Harry M. Hart, Chicago, and W. H. Driscoll, Syracuse, N. Y.

Dr. J. C. Geiger, director of Public Health for the city of San Francisco, was then introduced by Chairman Fleisher, and he presented the greetings of Mayor Angelo J. Rossi, and assured the assembly that San Francisco was very happy to entertain the group that is doing so much for the comfort of America. The session adjourned at 12:45 p. m.

FIRST TECHNICAL SESSION—

Tuesday, June 17, 1:30 P.M.

The first technical session of the 1941 Semi-Annual Meeting was called to order on Tuesday at 1:30 p. m., June 17, by W. L. Fleisher, president of the Society. He invited G. M. Simonson, general chairman of the Committee on Arrangements and past president of the Golden Gate Chapter, to outline the arrangements for the meeting.

Mr. Simonson welcomed the members and guests to California and announced that it was gratifying to learn that the registration had already far surpassed expectations. He assured the members that it would be a great pleasure for the Golden Gate Chapter and the committee to make their visit an enjoyable one.

President Fleisher acknowledged the Society's appreciation of Mr. Simonson's words of welcome, and called for the first technical paper Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson (complete paper published in August 1941, ASHVE Journal Section, *Heating, Piping and Air Conditioning*, p. 512). Professor Raber, professor of mechanical engineering, University of California, in presenting the paper mentioned that during the last decade and in happier world conditions than those existing today, engineering visitors to England have returned with intriguing reports of panel heating and cooling. Even large concrete office buildings were found, wherein human comfort was maintained by merely heating or cooling portions of the ceilings. In this manner, the engineering visitors reported that a large radiant energy effect upon the occupants was realized. Also, a decidedly lower temperature of air than 70 deg was used in the heating season. The reports also told of the definite impression of the invigorating quality of the indoor air under

the conditions described. Professor Raber then mentioned that Mr. Hutchinson, the co-author of the paper, spent four months in Europe studying English and panel systems in other countries during his stay, and he was one of those who returned with such reports.

Professor Raber further explained that the object of the paper was to present a basic rational analysis of the problem from which it might be possible to derive a simplified yet accurate proximate solution to facilitate design of radiant panel systems of higher effectiveness and less cost than by purely empirical methods. Research in panel heating based on the problems brought out in the paper, and leading toward a simplified method of design, he stated, is being conducted at the University of California in cooperation with the Society's Research Laboratory.

President Fleisher thanked Professor Raber for his interesting presentation and then stated that the importance of panel heating may be great or it may not, depending entirely on the point of view. Nevertheless, it is receiving more publicity today than practically any other method of heating. He brought out the fact that the idea of panel heating could be traced to the first century and was just coming into its own again. Following these remarks, President Fleisher called for discussion from the floor. Dr. F. C. Houghten, director of the Society's Research Laboratory, Pittsburgh, expressed his views on this very important and timely subject, and pointed out that this is the type of paper that usually results from intensive study at the Laboratory, coming from it with a wealth of mathematical analysis and is a means to an end and not an end in itself. Dr. Houghten discussed the paper in detail, concluding his remarks by stating that he felt that one of the most important things needed in this study at the present time was to include data in THE GUIDE which will allow one to easily determine the heat emissions from panels located in the ceiling, side wall or floor.

S. R. Lewis, Chicago, Ill., then offered some interesting discussion, citing as examples the systems in several houses he designed which have been in successful operation for three or four years. Hot water coils were embedded in the plaster of the ceiling, he said, but he also warned that the character of the construction should be studied carefully to determine the amount of surface required and what temperature that surface should be. He concluded his remarks by stating that people could live in comfort in houses in which some of the rooms have panel heating and some do not.

Dr. F. E. Giesecke, College Station, Tex., and past president of the ASHVE, expressed his opinion that panel heating is a very broad subject and that there is no fundamental difference between what is called panel warming and the ordinary type of heating. In other words, if cast-iron radiators are installed in a room heated by steam or hot water the air would be heated, and the air in turn heats the ceiling and the walls and they again radiate. The only difference in a room heated with ordinary radiators and one heated with panels is that there is perhaps a larger proportion of the heat transfer by convection with a radiator than with a panel warming system. In both systems heat is transferred by radiation and convection and the difference is only a variation in temperature. Dr. Giesecke pointed out that this subject should be treated in a broad general way and it must be remembered that in heating panels the entire room is filled with heat radiation to which the human bodies react, and in this case the bodies have a higher temperature than the exchanges from the bodies to the surroundings.

Prof. L. M. K. Boelter, Berkeley, Calif., was the last to discuss this timely topic. He referred to the mention made that panel heating was developed during the first century, and stated that the dark ages of panel heating, alluded to by Dr. Giesecke, should have also been mentioned. He further explained that any high-ceiling room behaves in a manner almost identical with a panel heated room. The hot air in the room, however it may be heated, will rise to the ceiling; the ceiling is 15 or 20 deg warmer than the floor, and a large fraction of the interchange between the system and the surroundings will be by radiation. Thus, the proponents of panel heating are merely adapting a heating system which operated quite satisfactorily in high-ceiling rooms to the present vogue of the low-ceiling room.

Mr. Hutchinson had no further comments to offer in closing the discussion, and President Fleisher expressed appreciation to Professor Raber and Mr. Hutchinson, and called for the next paper.

J. Donald Kroeker, Portland, Ore., presented his paper, Weather Data for Cooling Design Application (complete paper published in July, 1941, ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*).

With E. O. Eastwood, Seattle, Wash., first vice-president of the Society, presiding, discussions were called for, and John James, technical secretary of the Society, expressed the appreciation of the Society to Mr. Kroeker for reporting the local investigations presented in his paper, stating that this paper will benefit those who are responsible for preparing tables for the United States that are published in THE GUIDE. Mr. James further explained that the table now in THE GUIDE for design dry- and wet-bulb temperatures throughout the country has been developed as a result of the tabulations of a great number of figures obtained from a large number of sources, and that the problem now arises of correcting this table based on more recent information such as the type presented by Mr. Kroeker.

Professor Boelter offered his comments, stating that the collection of weather data through the United States mentioned by the author is a very important part of the air conditioning and heating program. In conclusion, Professor Boelter stated that he hoped that data on nocturnal irradiation as well as dry-bulb, wet-bulb and solar irradiation will be made available in more stations throughout the United States.

Chairman Eastwood thanked Mr. Kroeker for his presentation and called upon Noel E. Porter, Sacramento, Calif., who presented his paper entitled Concerning Conservation of Underground Water with Suggestions for Control (complete paper published in June 1941 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning.*)

Glen Miller, Los Angeles, Calif., gave an illustration of a recent installation when using well water, describing the method of control, which gave constant operation without starting and stopping, and which, he pointed

out, is apparently going to be a very satisfactory mechanical solution to the problem of water control.

President Fleisher resumed the chair and A. C. Buensod referred to comments by President Fleisher regarding an experience with a well, and related a similar experience in New York, questioning as to where all the water comes from in California and on the West Coast. He pointed out that in New York there are conservation rules which mean that you just cannot use the water; you have to go too far away to get it.

Mr. Porter joined in the discussion here and was greatly interested in Mr. Miller's remarks in regard to the method of control. He then answered Mr. Buensod's question as to where the water comes from, stating that the high Sierras, which are snowcapped through the year, feed the entire valley. He also mentioned that he felt that there would be sufficient water for many years to come.

John Kelly, Bakersfield, Calif., referring further to Mr. Buensod's question, mentioned that 20 or 30 years ago one used to be able to pump the water with a centrifugal pump; now it is being lifted with deep-well turbines set from 100 to 600 ft. When they go down much farther, he stated, these dams that are being built today will not do much good and the country may have to be given to the Indians.

Dr. B. M. Woods, University of California, Berkeley, Calif., was called upon by President Fleisher, and in commenting on the paper, remarked that the problem of water conservation has to be approached in several ways. He referred to Mr. Porter's discussion which applied to the part of the State in which the mean annual temperature, which is approximately that of the well water, is low enough to make the well water effective. The paper, he pointed out, only applies to those regions in which there is ample water and it should be emphasized that out in the West, water supply is a serious matter. On the West Coast weak salt solutions are encountered and sometimes they are not so weak, consequently the question of how to handle the water becomes a ticklish one. Then condenser problems arise and keeping the condenser clean in water that has a fairly high salt content is the problem. In certain regions where the water supply is low the cooling tower question comes up, and the cooling towers are matters of interest on this account.

Dr. Woods in concluding his discussion commented on one question which he felt was apparent. One must submit the control of the system and solve the problems of conservation, at the same time taking them largely out of the hands of human operation. Automatic control is rapidly coming and one can economize from it. He further stated that one must recognize that the greatest characteristic of the age in the mechanical field is the fact that one can run as many things as possible with the skill built into the machine and not into the operator and that applies all the way along the line. The small air conditioning plant, he pointed out, depends upon not paying very much to run it and that means electricity or internal combustion engine power at 25 to 35 cents per horsepower hour and is a good deal better than manpower at $400.00 per horsepower day.

President Fleisher then called upon Glen Miller to present his paper entitled Recent Developments in Absorption Refrigeration (complete paper to be published in the JOURNAL SECTION, *Heating, Piping and Air Conditioning*). Mr. Miller stated that in any discussion of a new development or the bringing up to date of an old development it seems that the thought given to the particular problem has to go through two or three phases; it must first go through a purely qualitative phase in which the general outline of the problem is worked out and the possibilities are outlined. The next stage is the problem of quantitative study of the things that have been found out in the qualitative approach, and then lastly a careful analytical demonstration of the entire problem. The author spoke on the first phase, the purely qualitative phase and explained that developments in absorption refrigeration are so rapid that no one can predict how far they are going, or to what use they will be put.

Mr. Miller's method of presentation was received with great interest, and at the conclusion of his paper, President Fleisher called for discussion from the floor.

C. F. Boester, St. Louis, Mo., expressed his opinion that as far as absorption refrigeration is concerned it has a very definite place in the domestic field, which he pointed out, is not necessarily limited to absorption refrigeration. Mr. Boester congratulated Mr. Miller on his analysis, and in conclusion brought out the point mentioned by the author, that the domestic field is very close and attractive to the heat and refrigeration experts because they can take advantage of a year-round load and with a very simply controlled switch for either summer cooling or winter heating.

President Fleisher then announced that amendments to the By-Laws of the Society would be read by A. V. Hutchinson, secretary of the Society. These amendments were presented in accordance with the terms of Article B-XVI, in a form submitted by the Committee on Constitution and By-Laws as follows:

Amendments to the By-Laws

Article B-III—Section 7—When the Admission and Advancement Committee has acted favorably upon a Candidate's application and assigned his grade the Council shall vote upon the election of the proposed Candidate for membership by letter ballot.

To be Amended as follows: When the Admission and Advancement Committee has acted upon a candidate's application and recommended his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot.

Article B-IV—Section 1—The admission fee of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1942 and thereafter the admission fee of Members and Associate Members shall be fifteen dollars ($15.00); of Junior Members five dollars ($5.00); and of Student Members two dollars ($2.00). Admission fee must accompany application.

To be Amended as follows: The Admission fee of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1945 and thereafter the admission fee of Members and Associate Members shall be fifteen dollars ($15.00); of Junior Members five dollars ($5.00); and of Student Members two dollars ($2.00). Admission fee must accompany application.

Article B-IV—Section 2—The annual dues of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1942 and thereafter the dues of Members and Associate Members shall be twenty-five dollars ($25.00); of Junior Members twelve dollars ($12.00); and of Student Members five dollars ($5.00).

To be Amended as follows: The annual dues of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1945 and thereafter the

dues of Members and Associate Members shall be twenty-five dollars ($25.00); of Junior Members twelve dollars ($12.00); and of Student Members five dollars ($5.00).

Article B-VIII—Section 7—There shall be an Admission and Advancement Committee of three (3) members, appointed by the Council, for a term of three (3) years, the term of one member expiring each year. This Committee shall receive from the secretary all applications for membership, make rigid inquiry as to the eligibility of candidates, and report to the Council **only such as have been approved.** In case of disapproval, only the proposers and the applicant shall be notified of such action. The proceedings of the Committee shall be private and confidential. It shall be the duty of this Committee each year to recommend to the Council, for transfer of grade of membership all Junior Members who have reached the age of thirty (30).

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The four amendments endorsed by the Council were voted upon separately and duly carried by unanimous vote.

President Fleisher before adjourning the session, introduced J. E. McNevin, president of the *Heating, Piping and Air Conditioning Contractors National Association,* holding its 52nd Annual Convention at the St. Francis Hotel during the same week. Mr. McNevin acknowledged the introduction and extended an invitation to all to attend the meetings at the St. Francis. Mr. McNevin was glad of the opportunity of being present and hoped to be able to get back to more of the technical sessions.

President Fleisher asked for further business and then announced that the meeting was adjourned at 4:15 p. m.

SECOND TECHNICAL SESSION—

Wednesday, June 18, 9:30 A.M.

President Fleisher called the meeting to order and the first paper, The Application of Gas Fuel to Existing Heating Boilers, by R. L. Grutzmacher (complete paper to be published in the ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*), was presented by the author. President Fleisher and E. K. Campbell, Kansas City, Mo., led in the discussion. President Fleisher thanked Mr. Grutzmacher for his presentation, and called for the next paper.

J. D. MacLean, Madison, Wis., responded and delivered his paper, Thermal Conductivity of Wood (complete paper published in the June 1941 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*). Mr. MacLean explained that the purpose of the experiments described in the paper was to make a comprehensive study of the factors that affect the conductivity of heat through wood. He further explained that the principal work was confined to conductivity in the radial and tangential directions; in other words, the right angles to the wood fibers, and with this in mind the tests designed to cover a large number of species have had a wide range of moisture content and a considerable variation in density within each species.

B. W. Farnes, Portland, Ore., extended a cordial invitation on behalf of the Oregon Chapter to the visiting members and their families to be their guests for a day or so on their return trips, to visit Bonneville Dam, Mount Hood, and take the Columbia River Highway trip. The trip included the large power plant at Bonneville Dam, the fish hatchery and the fish ladders which assist the salmon in getting up to the spawning ground, something which they felt the folks from the East were not familiar with and would be interested in seeing. Many of the members showed interest in Mr. Farnes' announcement, and President Fleisher was among those who hoped to enjoy the Oregon Chapter's hospitality.

For the benefit of the members returning via Los Angeles, Mr. Hutchinson announced that word had been received that the Southern California Chapter offered its services to those interested in visiting the movie studios of Warner Bros. or Twentieth Century Fox. H. H. Douglas, past president of the Chapter, A. J. Hess, and Leo Hungerford, president and secretary of the Chapter, were ready to welcome the members as they arrived.

President Fleisher then announced the next paper, The Effect of Insulation on Plant Performance in the Research Residence, by A. P. Kratz and S. Konzo (complete paper published in the May 1941 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*), which was presented by Professor Kratz. He pointed out that the tests discussed in the paper were the results obtained from work done in cooperation with the University of Illinois and the *National Warm Air Heating and Air Conditioning Association,* which association built the Research Residence at the University and maintained it since 1924. Professor Kratz mentioned that the residence was primarily for the purpose of running tests on warm air furnaces and heating systems, and further stated that the Society has at times entered into triangular cooperation on summer cooling, but the work discussed in this paper is all winter heating.

Professor Eastwood, first vice-president of the Society, then took the chair and called for discussion from the floor. O. W. Johnson, San Francisco, responded and questioned whether the discrepancy between the calculated and observed wall temperature would be explained on the basis of assumed film factors by the effect of the paper or something to do with the surface itself.

Professor Kratz explained that the heat loss factors are going to be an assumed thing, but they are stated under one condition only. In THE ASHVE GUIDE they are estimated under conditions in which you have a 40 deg mean temperature and a 15-mile wind. He further brought out that the variations in either one of those will affect the value of heat loss coefficients and affect the temperature you would get as compared with the calculated temperature, and would affect the calculated heat loss as compared with the actual heat loss. Undoubtedly variations in the film are the largest variations that enter into that picture.

President Fleisher brought up the question of the sun effect, and Dr. Giesecke expressed his opinion that the Society should give special thought to the film coefficient which involves radiation and convection.

In replying to Dr. Giesecke's comment, Professor

Kratz stated that film coefficients are based entirely on the conception that the objects inside the room or outside, wherever they are, are at the same temperature as the air, and if any surfaces are either warmer or colder than the air temperature in the neighborhood of the surface, the film coefficient given in THE GUIDE does not hold at all because the relative proportions of convection and conduction have changed.

Cecil Jones, Bakersfield, Calif., also offered his comment as to what was the effect on the human being in the room when it was on the wall, say 15 deg lower in temperature.

H. H. Douglas, Los Angeles, Calif., cited his experiences in the past eight years in which 32 homes were built scattered over Southern California, half were insulated and the remaining uninsulated. Therefore, the kilowatt hour input was juggled about and it was found that between the two different groups there was a difference in cost of operation of exactly 45 per cent, which confirms Professor Kratz's figures. In contrast to the Research Residence, he pointed out that the houses in California were not built as a cube or block; sometimes "U" shaped with two or three or more different wings and angles, and consequently the outside exposure, which is the determining factor on heat losses is tremendously large in proportion to the volume as compared to the Research Residence. Professor Eastwood thanked Mr. Douglas for his interesting discussion, and following additional remarks by Professor Kratz, turned the meeting over to President Fleisher.

S. H. Downs, Kalamazoo, Mich., brought up a question about the comparison of the total heat loss, to which Professor Kratz replied.

G. L. Tuve, Cleveland, O., brought up the question of the measurement of temperature in rooms in the Research Residence, in which he felt there was very little difference in the effect on the floor to ceiling temperatures with the insulated house and he seemed surprised to think that people would be comfortable. Professor Tuve was concerned about the method of measurement, temperature measured at the middle only and with thermometers or thermocouples, and with a set of measuring devices in each room.

Professor Kratz was glad that Professor Tuve opened up this broad subject, but replied briefly that all the temperatures in the Research Residence were measured during the heating season by using the temperature on which the indoor-outdoor temperatures were based. All the curves are based on readings of thermometers in the central axis of the room, Professor Kratz explained, and an average of 11 rooms so measured.

S. R. Lewis mentioned that he was interested in insulation as applied to the cooling because all of that as far as he had gathered had been on the basis of the hot water or steam as used in panel heating. Mr. Lewis then quoted Dr. A. C. Willard, president of the University of Illinois, a few years back, who said, "if you increase the efficiency of the insulation in the ceiling, thereby whatever is above that ceiling is going to be much colder and the temperature difference between what is below the insulation and that which is above the insulation in the attic is going to increase." Mr. Lewis felt that where one ends and the other begins is rather difficult to say, and it may be the more insulation you put on the more trouble you run into in some cases, although that is not quite true. He brought out the fact that he and Dr. Giesecke had made some study of the drafts, downflowing currents along the windows in rooms that are thoroughly insulated and are heated with 130 deg plaster in the ceiling.

Mr. Douglas then brought up the question in connection with the fly-wheel effect or matter of storage involved in California and the South Coast country. The changes mentioned are so great between day and night that the flywheel effect becomes very apparent. He related an interesting example which he had an opportunity to study some years ago at California Tech, in a research study on insulations and various insulation tests. A man he knew became so interested in that study that when he built his new home in Pasadena he used 8 in. studding and filled the walls with redwood sawdust, which virtually made an ice-box out of the home. The owner stated that his home is 10 or 15 deg cooler in summer and easier to heat in the winter time as it used just half of the fuel used previously.

President Fleisher again called upon Professor Kratz to present the paper, An Analysis of Factors Influencing Building Heat Losses, by Paul D. Close (complete paper to be published in the ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*). Professor Kratz in presenting a summary of the paper stated that the author analyzed the sources of variations in heat loss calculations, showing why deviations occur, both in the actual as compared with calculated and between the different calculated values. The author also takes into account the heat capacities of the building.

President Fleisher thanked Professor Kratz for presenting Mr. Close's paper and congratulated him for the manner in which he presented it. Following several announcements concerning the next technical session the meeting adjourned at 12:15 p. m.

THIRD TECHNICAL SESSION—

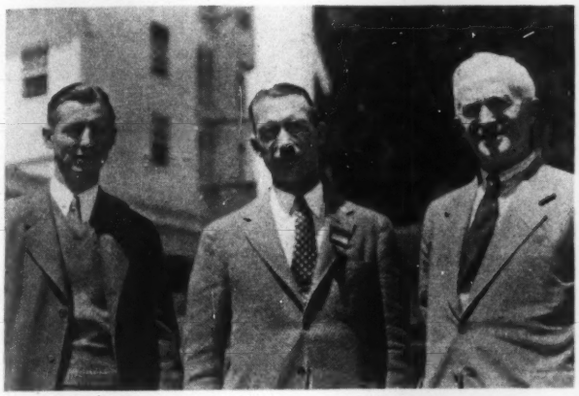
Thursday, June 19, 9:30 A.M.

The third technical session was called to order by President Fleisher, and the first paper, Resume of Society Research in Cooperative Institutions, which was to have been delivered by A. E. Stacey, Jr., chairman of the Committee on Research, was presented by Mr. James. Twenty-four years ago the idea of fundamental research on heating, ventilating and air conditioning problems was suggested to the Society, as related by Mr. James, and in 1919 the ASHVE Research Laboratory, located in the U. S. Bureau of Mines, Pittsburgh, Pa., now under the direction of Dr. F. C. Houghten, became a reality. Since that time, he explained, over $600,000 has been spent on a variety of subjects of which approximately 50 per cent of that amount has been received from membership dues and the balance has been contributed by various manufacturers and trade associations.

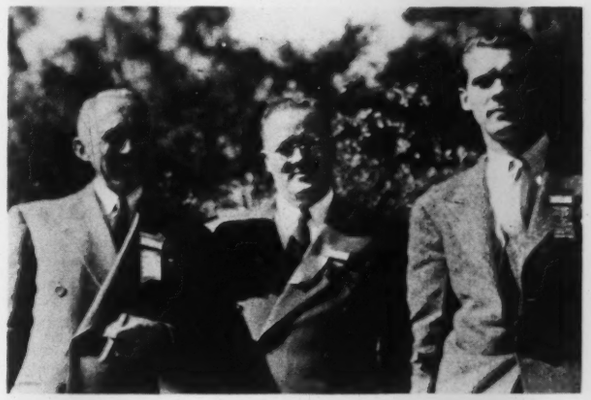
Mr. James also stated that in addition to maintaining its own Research Laboratory in Pittsburgh, the Committee on Research at the present time is actively sponsoring cooperative research in 13 institutions. At the conclusion of Mr. James' presentation, President Fleisher expressed his gratitude to him for presenting the activi-

Pres. W. L. Fleisher, New York, and Dr. B. M. Woods, Berkeley

Members of Council—A. P. Kratz, Urbana, S. H. Downs, Kalamazoo

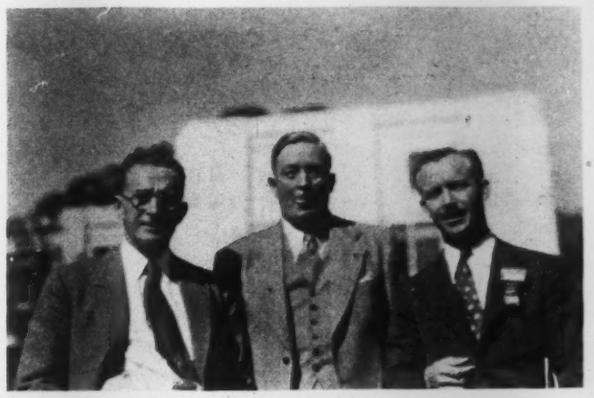


From Chicago—M. W. Bishop, C. E. Price and S. R. Lewis



H. H. Douglas, H. H. Bullock, J. B. Griffith, Southern California members

Pres. C. E. Bentley of Golden Gate Chapter, Mrs. Bentley, R. A. Folsom



Pacific Northwest Chapter Officers, R. E. Chase, *Treasurer*, F. J. Pratt, *President*, H. T. Griffith, *Vice-President*

From Portland—J. J. Byrne, W. R. Burton, Harry Gribbon

DeWitt Clarke, Treasurer Merrill F. Blankin and R. L. Blanding

A group of ASHVE members at Berkeley

ties of the Society in such an excellent manner, and S. R. Lewis joined in thanking Mr. James and suggested that a message be sent to Mr. Stacey expressing a vote of thanks for the progress being accomplished by the Committee on Research under his direction.

The Interaction Constant for Moist Air, by John A. Goff and A. C. Bates (complete paper published in the July 1941 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*), was the next paper, and was presented by Prof. L. M. K. Boelter, University of California. Professor Boelter announced that he was requested by Dean Goff to take his place in putting this material before the meeting. He then attempted to give his points of view with respect to the significance of this research. He brought out that the authors had proceeded by extremely ingenious algebra and then later by experimental devices, to measure the deviation from Dalton's Law, making use of thermodynamics and the Kinetic theory. By using this algebraic manipulation the authors show that it is possible for them to determine the so-called interaction constant of molecules of water vapor and air by an apparatus which they illustrated in the paper. Professor Boelter also referred to the author's mention of experimental data of German experimenters published in 1924, which show that the intermolecular constant, or something proportionate to it, varies somewhat. In concluding, Professor Boelter felt that the Committee on Research should be congratulated for undertaking a basic research of this character and leading the way in the determination of some of the deviations from perfectness, which Dean Goff and Mr. Bates brought out.

President Fleisher expressed appreciation to Dean Goff, who is so eminent in thermodynamics, for developing the theory and the paper Professor Boelter presented so well.

The next paper, Local Cooling of Workers in Hot Industry, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet (complete paper published in the July 1941 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*) was given by Dr. Houghten. In this paper it was brought out that this investigation has developed something very interesting so far as industry is concerned in the emergency which exists at the present time, where the owner, manufacturer and the government itself is quite undetermined as to what conditions they should provide to properly manufacture the goods required in the emergency. For the last few years investigations have been conducted to find the maximum conditions under which people can work and produce the goods economically in all of these so-called black-out air conditioned buildings. Dr. Houghten brought out that one of the conditions observed in this investigation was the condition of perspiration of the worker. The worker without the cooling process in many cases was wringing wet and his clothing was wet throughout, while the worker who was provided with the cooling system did not perspire noticeably excepting at very high temperatures. The paper mentions temperatures at which perspiration begins at different effective temperatures, but you can entirely control perspiration by cooling with either of these methods, Dr. Houghten explained. The hotter the ambient condition the cooler the air, the lower the temperature at which the air must be supplied. S. H. Downs commented on the difference in the temperature required for the air in the suit and in the blast where the man is working.

The paper, Economical Air Velocities for Mechanical Air Filtration, by Frank B. Rowley and Richard C. Jordan (complete paper published in August 1941 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*) was presented by Mr. Jordan. Mr. Jordan explained that this paper was the result of research sponsored by the Society in cooperation with the Engineering Experiment Station of the University of Minnesota. President Fleisher resumed the chair, and expressed his appreciation to Professor Rowley and Mr. Jordan for this paper of special interest and importance. He extended an invitation to all members to send in written discussion on this paper.

M. F. Blankin, Philadelphia, Pa., treasurer of the Society and chairman of the Resolutions Committee, was then called upon and presented the following resolutions prepared by the Committee, which were voted upon and unanimously carried:

Resolutions

WHEREAS, the members of the ASHVE who are attending the First Pacific Coast Meeting of the Society desire to express their sincere appreciation for the splendid program which has been presented, now therefore,

BE IT RESOLVED, that an expression of thanks and appreciation be adopted and spread upon the minutes of the Society and copies thereof be transmitted to each of the following persons and agencies that have contributed to make this meeting so enjoyable for the members of the Society who attended:

To the Golden Gate Chapter of the Society and their ladies who have received us with such gracious hospitality;

Also to the three other Pacific Coast Chapters who have cooperated so well with the Golden Gate Chapter;

To the members of the Committee on Arrangements for the fine program of entertainment;

To the authors of the technical papers for their valuable contributions and discussions;

To Mayor Angelo J. Rossi for his assistance in officially opening the Exposition and the many courtesies which he extended to us;

To the Management and Staff of the Palace Hotel for their excellent service;

To Mr. Walter Swanson and the staff of the San Francisco Convention and Tourist Bureau for their fine cooperation and assistance;

To the newspapers for their coverage of our sessions and to the trade papers for their excellent publicity and attendance at the meetings;

To the railroads and airlines for their courtesies and efficient service;

To the Postal Telegraph Company for its courtesy in furnishing the registration service board;

To the University of California for arranging for the inspection trip and meeting at the Engineering Building and the tea at International House;

To Dr. Chauncey D. Leake for his splendid talk at the Semi-Annual Banquet;

To Dr. B. M. Woods for the excellent job he did as Toastmaster at the Semi-Annual Banquet;

To Mr. C. F. Roth and his associates for arranging a splendid exposition and to the exhibitors for their participation and cooperation,

And finally to recognize A. V. Hutchinson and his eighteen years of service in the Society and his fifteen years of continuous service as Secretary.

RESPECTFULLY SUBMITTED,

M. F. BLANKIN, *Chairman;* M. W. BISHOP, R. L. BLANDING.

Clyde E. Bentley responded to Mr. Blankin and expressed the appreciation of the Golden Gate Chapter for the opportunity of acting as host to the Society.

President Fleisher then announced that the last technical session would convene at the University of California, Berkeley, at 2:00 p. m. The session adjourned at 11:45 a. m.

FOURTH TECHNICAL SESSION—

Thursday, June 19, 2:00 P.M.

On Thursday, June 19, President Fleisher called the final technical session to order at 2:00 p. m. in Room 104 of the Engineering Building, University of California. He introduced Prof. L. M. K. Boelter, who presented Part I of the paper Performance Characteristics of a Forced Draft, Counterflow Spray Cooling Tower, by Howard H. Niederman, E. D. Howe, John P. Longwell, Ralph A. Seban and L. M. K. Boelter (complete paper to be published in the ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*). This was followed by the presentation of Part II, which was given by H. B. Nottage, Berkeley. The paper was illustrated with slides and presented an analysis of the research work conducted on various types of cooling towers.

Entertainment

The Committee on Arrangements under the able leadership of G. M. Simonson had arranged an elaborate entertainment program for visiting members and everyone who attended the five-day meeting retains many pleasant memories of his visit with the Pacific Coast members.

The first get-together was the joint luncheon of engineers and contractors at the St. Francis Hotel on Monday, after which Presidents Fleisher and McNevin and Honorable Angelo J. Rossi, mayor of San Francisco, officially opened the Pacific Heating and Air Conditioning Exposition at the Civic Auditorium.

At 2:00 p. m., a golf tournament was held at the Lakeside Country Club and those who participated played the ocean course. The Research Cup was awarded to James Gayner, San Francisco, with a score of 86-15-71. Other awards were made to C. P. Hensley and Fred W. Kolb, San Francisco. In the competition for the Eichberg Memorial Cup the Golden Gate Chapter team had the best score; James Gayner, 86-15-71; W. P. Scott, Jr., 86-9-77; Fred W. Kolb, 102-24-78. The runner-up was the Pacific Northwest Chapter team; George E. Mead, 87-6-81; H. K. Mead, 101-20-81; and H. T. Griffith, 111-20-91. A special guest prize was awarded to E. J. Balaam with a low gross score of 81.

Many of the Society members attended the annual dinner and dance of the *HPACCNA* at the St. Francis Hotel at 7:00 p. m. on Monday evening and enjoyed an excellent meal, a variety program of entertainment and dancing in the ballroom.

On Tuesday morning the ladies left the Palace Hotel in sightseeing buses for a trip over the Golden Gate Bridge to Marin County and the Muir Woods, and Mt. Tamalpais. This was a thrilling trip through the redwood forest, with an enjoyable luncheon served at the Muir Woods Inn. J. F. Kooistra, chairman of the Ladies Committee, was in charge of arrangements and his committee members served as guides on the trip.

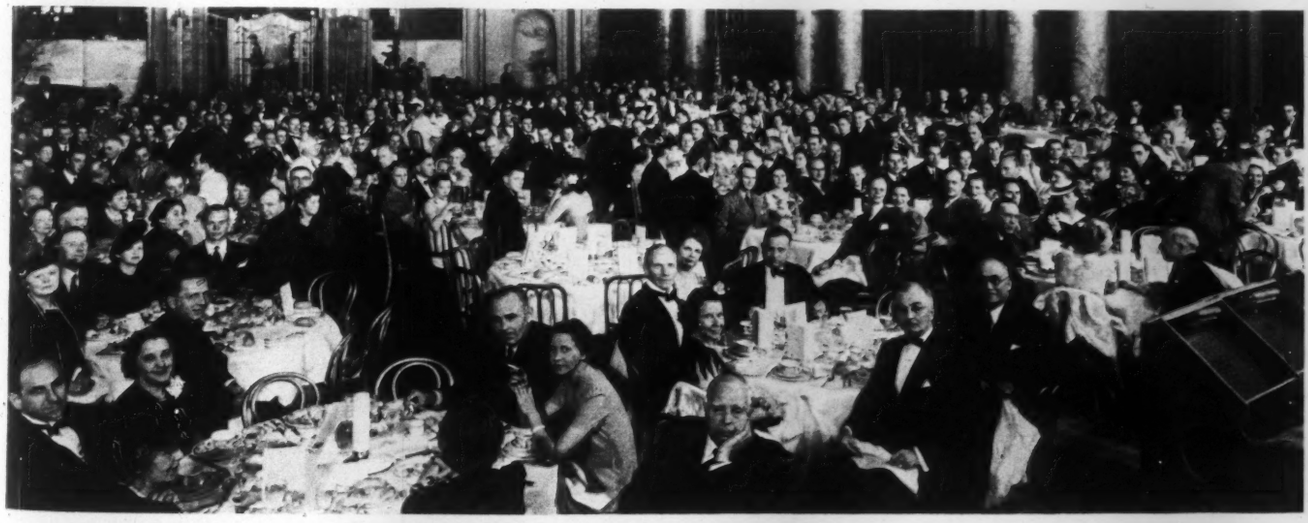
During the evening there was a sightseeing trip through Chinatown and to some of the historic landmarks of San Francisco. Great interest was evidenced in the Chinese telephone exchange, Chinese theater and night club, as well as the variety shops.

On Wednesday morning the ladies left the Palace Hotel for a scenic ride through San Francisco, including a visit to Seal Rocks, Golden Gate Park and the Civic Center, after which luncheon was enjoyed at the famous Cliff House.

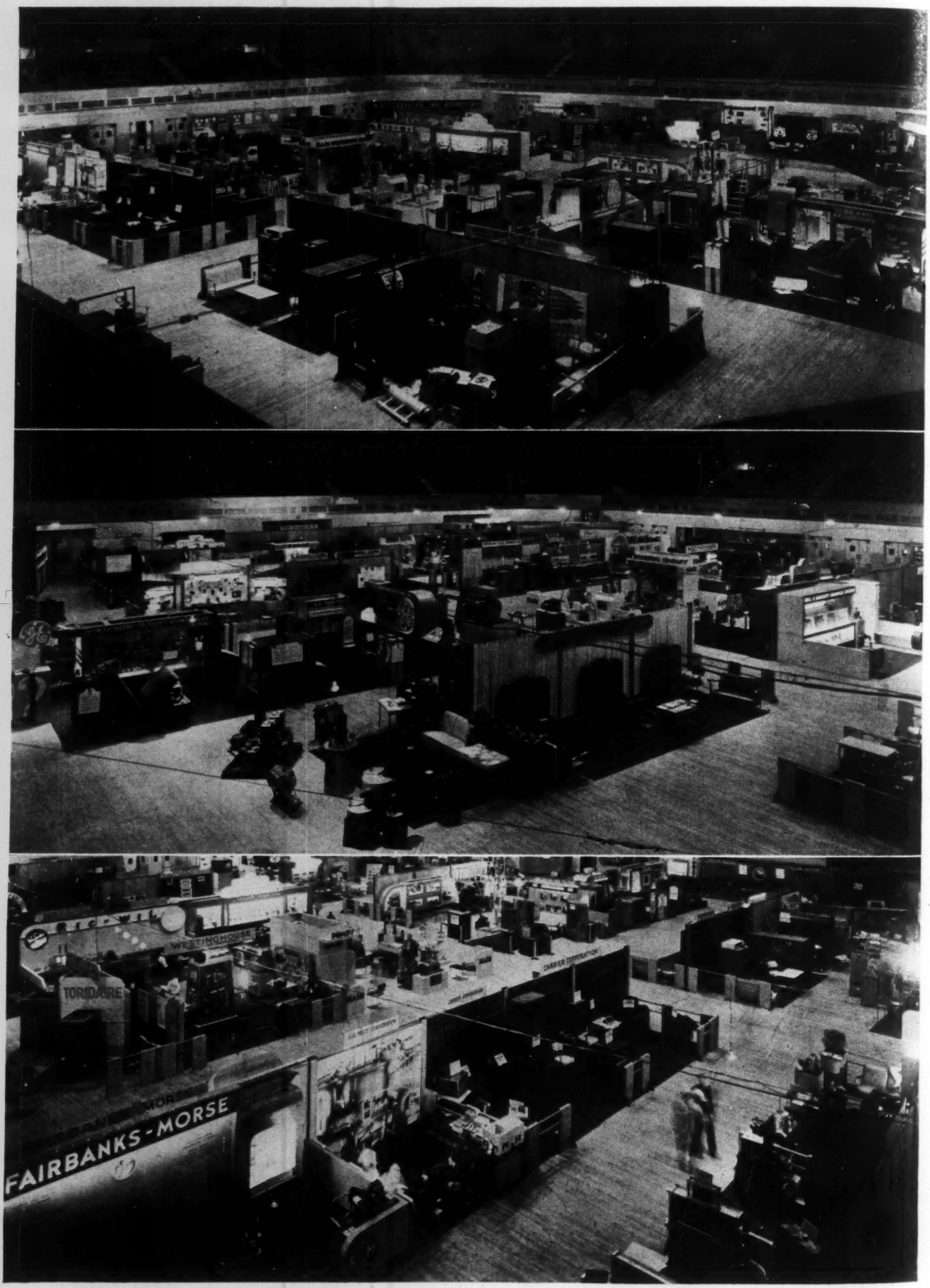
The Society's Semi-Annual banquet and dance was held in the Palm Court of the Palace Hotel on Wednesday evening at 7:30 p. m., and after an excellent dinner, the toastmaster, Dr. B. M. Woods, presented President W. L. Fleisher, who introduced the distinguished guests at the speakers' table. M. F. Blankin presented the Research Cup in the absence of the chairman of the Committee on Research, A. E. Stacey, Jr. Mr. Blankin also presented the Eichberg Memorial Cup to the Golden Gate Chapter.

Prof. E. O. Eastwood, First Vice-President of the Society, presented the Past President's Memory Book on behalf of the Council to Dr. F. E. Giesecke, who responded with a brief acknowledgment of his appreciation.

Dr. Woods then presented the speaker of the evening, Dr. C. D. Leake, professor of pharmacology at the University of California Medical School, who gave an interesting address on Human Engineering. The toastmaster then invited the local chapter officers and the members of the Committee on Arrangements of the Golden Gate Chapter to acknowledge their introduction to the audience. Dr. Woods introduced the secretary of the Society and read a resolution presented by the Nominating Committee calling attention to the fact that A. V. Hutchinson has completed 15 years service to the Society as Secretary.



Semi-Annual Banquet of ASHVE, Palace Hotel, San Francisco



General view of exhibits at Pacific Heating and Air Conditioning Exposition

Following dinner an entertainment program was presented in an adjoining room with Armand Girard as master of ceremonies, after which dancing was enjoyed in the Palm Court to the music of Larry Cannon and his orchestra.

At noon on Thursday members and ladies rode from the Palace Hotel over the San Francisco-Oakland Bay Bridge to Berkeley where luncheon was served at the Claremont Hotel to 75 members and ladies. The group was welcomed by Dr. Woods, and Professor Raber gave a brief outline of the university's history and described some of the points of interest on the campus. After lunch the members attended a technical session in the Engineering Building, while the ladies enjoyed a sightseeing trip around the campus, after which tea was served at International House through the courtesy of the University of California. An inspection of the University laboratories was enjoyed by the group of 200 engineers and contractors, with the engineering department staff acting as guides.

On Friday a joint program was participated in by the members of the ASHVE and the *HPACCNA* members, commencing with an inspection trip at 9:30 a. m., to Station "A" of the Pacific Gas and Electric Co.

A party left the dock early in the morning for a deep sea fishing trip, while another group held a golf tournament at the Lakeside Country Club. Many of the members and ladies made an early morning shopping tour in the downtown district and in the novelty shops of Chinatown.

Exposition

When the master switch was pulled on June 20 the great mechanical exhibits at the Pacific Heating and Air Conditioning Exposition ended their short but vibrant life. For five days one of the most important and comprehensive expositions ever held in San Francisco was visited by many thousands of visitors who came to wander, stayed to wonder. And in San Francisco where Nature is generous with its supply of natural heat and air conditioning, Californians and others from the 11 Western states were brought to a dazzling realization of what science and research has accomplished for human need and comfort.

Sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, the exposition was acclaimed as one of the greatest contributions to public enlightenment of industrial foresight and progress. According to Charles F. Roth, exposition manager, displays of heating and air conditioning units and equipment became a magnetic attraction to many visitors planning the construction of homes, as well as leading engineers and operators.

The exposition proved an added attraction to members and guests of the ASHVE and the *Heating, Piping and Air Conditioning Contractors National Association* whose meetings were held in San Francisco during the same period.

Organized visits were arranged by both groups several times during the week. Commenting on the value of the exposition, many of the Engineers' and Contractors' officers expressed appreciation of this cooperative venture which has directed national attention to the work and accomplishments of both organizations.

Among the 7,500 who registered were representatives of outstanding construction companies, steamship operators, contractors, engineers, industrial and commercial building owners and operators, home owners, architects, designers, dealers and distributors.

Products and equipment were displayed by 83 of the foremost manufacturers and designers in the country. Included among these exhibits were active displays of air conditioning equipment, boilers and furnaces, fans and blowers, control apparatus, registers and grilles, pumps, oil and gas burners, insulation material, piping, unit heaters, steam traps, steam specialties and radiators.

With defense activities approaching rip-tide velocity in California, local and out-of-state contractors evinced intense interest in the exposition. Representatives of many contractor and construction groups attended frequently to survey equipment on display. This interest was accepted as a general reflection of proposals to include modern and efficient heating and air conditioning equipment in all public and private construction projects planned for the immediate future.

Well attended by selected audiences drawn from every part of the United States, the exposition was declared a distinct success, both from the point of view of the exhibitor and appreciative visitors.

The exposition was arranged and managed by Charles F. Roth, President of the International Exposition Co., Grand Central Palace, New York.

Candidates for Membership

Unless objection is made by some member by August 15, 1941, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES

ATKINS, GEORGE E., Consulting Engr., San Francisco, Calif.

BARRETT, CAMPBELL M., Mgr., General Equipment Co., Ltd., Vancouver, Can.

BLAIR, ERNEST L., Industrial Engr., Stone & Webster Engrg. Corp., Boston, Mass.

BOLAND, L. C., JR., Consulting Engr., Atlanta, Ga.

COLLINS, LEO F., Chemist, The Detroit Edison Co., Detroit, Mich.

DICKENS, LESTER A., Partner, Dickens, Scheufler, Burens, Inc., Cleveland Heights, Ohio.

DONOHOE, CHARLES F., Engr., The Detroit Edison Co., Detroit, Mich.

DOVENER, ROBERT F., Engr., Atchison & Keller, Washington, D. C.

HARBIN, FRANK, JR., Engr., Home Furnace Co., Holland, Mich.

HOECKER, GEORGE F., JR., Sales Mgr., Plbg. & Htg., Sears, Roebuck & Co., Union City, N. J.

HOWES, EDWARD W., Dist. Repr., General Electric Co., San Francisco, Calif.

HUNTER, THOMAS B., Consulting Engr., Hunter & Hudson, San Francisco, Calif.

INGHAM, JOHN F., Pres.-Treas., John F. Ingham, Inc., Greenwich, Conn.

REFERENCES

Proposers	*Seconders*
R. A. Folsom	W. W. Cockins
N. H. Peterson	J. F. Kooistra
A. S. Leitch	E. A. Mead
E. A. Allcut	I. C. Jennings
J. W. Brinton	D. M. Archer
C. R. Swaney	E. G. Carrier
T. T. Tucker	A. H. Koch
C. L. Templin	L. F. Lawrence, Jr.
E. E. Dubry	G. D. Winans
J. H. Walker	S. S. Sanford
L. E. Slawson	L. T. Avery
E. W. Gray	C. M. H. Kaercher
J. H. Walker	J. N. Livermore
G. D. Winans	C. H. Bay
R. D. Terhune	L. F. Nordine
Thomas Baker	J. J. Nolan, Jr.
W. C. DeRoo	F. C. Warren
T. D. Stafford	H. J. Metzger
William Ritchie	J. L. Hodges (*ASME*)
W. G. Christy (*ASME*)	S. L. Boyar
C. D. Williams	N. H. Peterson
L. G. Trolese	C. E. Bentley
N. H. Peterson	J. F. Kooistra
W. W. Cockins	B. M. Woods
A. J. Offner	H. L. Alt
C. S. Koehler	J. R. Smak